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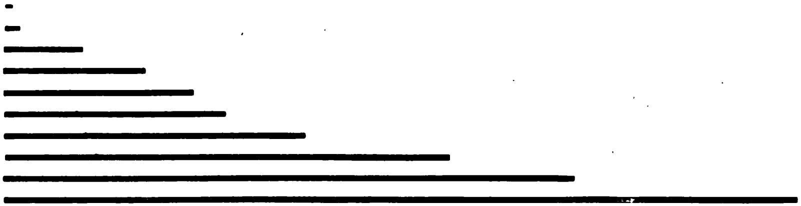
1887	9,281	49,844
1888	27,696	77,540
1889	26,065	103,605
1890	50,502	154,107



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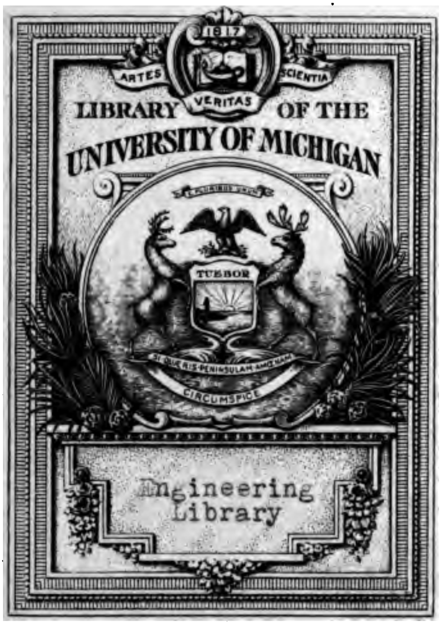

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COMPOUND LOCOMOTIVES.

BY

ARTHUR T. WOODS, M. M. E.,

PROFESSOR OF MECHANICAL ENGINEERING,
UNIVERSITY OF ILLINOIS.

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PREFACE.

In the preparation of the series of articles for the columns of the *National Car and Locomotive Builder* which are here collected in book form, the aim of the author was to combine the description of the various forms of compound locomotives which have been actually used, with so much of the theory of the design of compound engines as would seem to be directly applicable to locomotive practice.

An effort has been made to present an unprejudiced analysis of each type, and to point out such advantages and disadvantages as are apparently clearly demonstrable, while carefully avoiding matters of individual preference.

Free use has been made of all available material, and the authority for data is in general given in the text. The author wishes to specially acknowledge his indebtedness to *Engineering* and to Mr. Anatole Mallet, civil engineer, Paris; Mr. A. von Borries, locomotive superintendent of the Hanover Railroad; Messrs. Henry and Baudry, of the Paris, Lyons & Mediterranean Railway, and Mr. G. Du Bousquet, of the Northern Railway of France, for courteously supplying him with information concerning their designs.

CHAMPAIGN, Illinois, January, 1891.

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COMPOUND LOCOMOTIVES.

CHAPTER I.

The elementary theory of compound locomotives does not of course differ from that of other compound engines, whether they are condensing or non-condensing, or for stationary or marine purposes; but in applying the principles of compound engines to locomotives, while careful attention must be given to each of the many items which enter into the problem of designing any well-proportioned compound engine, it will be found that some factors, which are of comparatively small consequence in marine or stationary work, become of importance in the locomotive. These differences arise largely from the wide range of power required from locomotives, and the practical necessity of keeping the valve gear and operating mechanism as free from complication as possible. On the other hand, the recent introduction of higher pressures and greater piston speeds in marine practice has rendered some of the working conditions of marine engines and locomotives much more nearly alike than they have been heretofore.

ELEMENTARY INDICATOR CARDS.—The theory of the compound engine has been so thoroughly discussed by able writers, particularly in text books on the marine engine, that any further discussion seems almost superfluous; but as the most satisfactory method of designing any steam engine is to start with the elementary theoretical indicator card, and alter and adjust it here and there as experience and the special requirements of the case dictate, we will follow that method as far as practicable in considering compound locomotives. The type of compound locomotive which most nearly resembles the ordinary form of simple engine is the two-cylinder receiver engine having the cranks at right angles, the high-pressure crank leading. In this form, the steam, after expanding in the high-pressure cylinder, is exhausted into an intermediate receiver from which the low-pressure cylinder is supplied. We will first consider the form of the elementary theoretical indicator cards from such an engine, and for the present will assume that there are no clearance spaces, that steam is admitted and exhausted exactly at the beginning and end of the stroke respectively, that the point of cut-off is sharply defined, and that irregularity, caused by the angularity of the connecting rods, may be neglected. If with these assumptions steam is cut off at one-half stroke in each cylinder, the indicator cards would have the general form shown in Fig. 1.

The upper card, *a*, *b*, *c*, *d*, *e*, *f*, *a*, is from the high-pressure cylinder and the lower card, *e*, *f*, *g*, *h*, *k*, *e*, is from the low-pressure cylinder. The cards are shown as they would be traced by the pencil of an indicator, which is applied first to one cylinder and then to the other, the same spring being used and both cards having the same length. The two cards are then fitted together as shown in Fig. 1, so as to represent the successive actions of the steam in the two cylinders, the cranks being at right angles. Referring to the high-pressure card, *a b* is the admission line and *b c* the expansion line, as in a card from a simple engine. At *c* the high-pressure exhaust opens, connecting the high-pressure cylinder with the receiver, and the pressure falls to *d*. The conditions which control this drop in pressure *c d* will be discussed later. As we have assumed a cut-off at one-half stroke in each cylinder, the steam valve of the low-pressure cylinder closes just as the high-pressure piston reaches the end of the stroke at *c*. Hence, as the high-pressure piston makes the back stroke, the steam is compressed in the high-pressure cylinder and receiver until one-half the back stroke is accomplished at *e*. Here the low-pressure valve opens, admitting steam from the receiver to that cylinder, and from *e* to *f* there is free communication between the exhaust side of the high-pressure cylinder, the receiver, and the steam side of the low-pressure cylinder. At *f* the high-pressure exhaust valve and the low-pressure steam valve close, and the steam expands in the low-pressure cylinder to the end of the stroke *g*, when it is exhausted into the atmosphere as from a simple engine. The remaining lines of the cards do not differ from the corresponding lines in cards from simple engines.

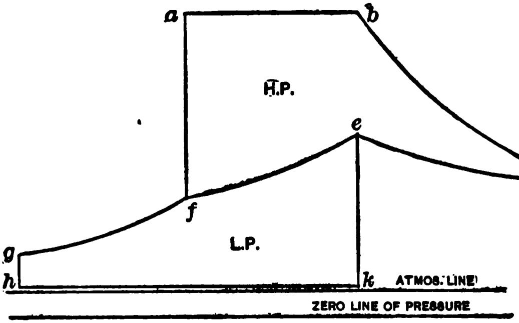

Fig. 1

Formulas for Calculating Pressures.—For calculating the pressures at the various points in the cards we can without serious error make use of the ordinary formulas for "moist steam," or in other words, assume that pressures vary inversely as the volumes, the curves of expansion and compression then being rectangular hyperbolas. On this basis mean pressures for such lines as *a b c* are determined by the formula $p_m = p_1 \frac{1 + hyp.\ log.\ r}{r}$ (1). This will be recognized as the ordinary formula for mean pressures, and in which p_1 is the absolute initial pressure, r is the ratio of expansion, and p_m is the gross mean forward pressure. The absolute pressure is the gauge pressure plus the atmospheric pressure, which is ordinarily taken as 14.7 pounds per square inch. Tables of hyperbolic logarithms or of values of the fraction $\frac{1 + hyp.\ log.\ r}{r}$ for various values of r are given in almost all books on the steam engine, and will therefore be omitted here. As an example of the application of the above formula, let the gauge pressure at the point *b* in Fig. 1 be 145.3 pounds per square inch, so that the absolute pressure will be 160 pounds. As cut-off takes place at one-half stroke the ratio of expansion $r = 2$, and therefore the final pressure in the high-pressure cylinder will be one-half of 160 = 80 pounds. From the tables previously referred to we find that for $r = 2$, $p_m = .847\ p_1$, and therefore $p_m = .847 \times 160 = 135.5$ pounds absolute pressure, which is the mean pressure between *a* and *c* measured from the zero line of pressures. This formula is applicable to such lines of the card as *a b c* when *a b* is parallel to the atmospheric line, as it is practically in engines supplied from a boiler and working at slow speeds. For calculating the mean pressure between *b* and *c*, *d* and *e*, *e* and *f*, or for other expansions or compressions in which the part of the card considered is wholly curved, or where no line of constant pressure as *a b* is included, the formula $p_m = p_1 \frac{hyp.\ log.\ r}{r - 1}$ (2) is to be used. This formula is deduced from the same theoretical considerations as the one given above, and the letters represent similar quantities. The former covers the whole stroke from *a* to *c*, and the latter only the curved part of the card as from *b* to *c*. For example, the pressure at b is 160 pounds and the volume at c is twice that at b. The ratio of expansion is therefore 2, and by reference to a table of hyperbolic logarithms we find $p_m = 160 \frac{.693}{2-1} = 160 \times .693 = 110.9$ pounds between b and c. This is for one-half of the stroke, and for the first half, from a to b, the mean pressure is 160 pounds, therefore the average for the whole stroke would be $\frac{160 + 110.9}{2} = 135.5$, which is the same as calculated by the first formula.

To avoid needless repetition, the following symbols, which will hereafter be used, are here inserted:

v = volume of high pressure (h. p.) cylinder in cubic inches.

V = volume of low pressure (l. p.) cylinder in cubic inches.

C = volume of receiver in cubic inches.

R = ratio of the two cylinders.

Pressure in the Receiver.—As before stated, there is in the case illustrated by Fig. 1, compression in the h. p. cylinder and receiver from d to e, and expansion in the h. p. cylinder, the receiver and the l. p. cylinder from e to f. How much the pressure varies between d, e and f depends upon the capacity of the receiver as compared with the h. p. and l. p. cylinders. In the present case, assume the capacity of the receiver to be equal to that of the h. p. cylinder, or $C = v$, and let the pressure at e be taken at 96 ponnds. At d the steam fills the h. p. cylinder + receiver, and at e it fills one-half the h. p. cylinder + receiver; therefore the compression is from $v + C = 2v$ to $.5v + C = 1.5v$, and the ratio of compression is $2v + 1.5v = 1.33$. The pressure at d is then $96 \times .75 = 72$ pounds, and by formula (2) the mean pressure between e and d is $96 \times .86 = 82.6$ pounds. At f the volume occupied is that of one-half the l. p. cylinder + receiver. Assuming for the present case that the l. p. cylinder is 2.5 times the h. p. cylinder, or $R = 2.5$, the expansion will be from 1.5 v to $\frac{2.5\,v}{2} + C = 2.25\,v$, or the ratio of expansion is $2.25\,v \div 1.5\,v = 1.5$. The pressure at f is then $96 \times .67 = 64$ pounds, and by formula (2) the mean pressure between e and f is $96 \times .81 = 77.8$ pounds. The following table shows the pressures at the points d, e and f of the h. p. indicator card with receivers having capacities of v, 1.5 v and 2 v, and with cylinder ratios of 2 and 2.5, which give a sufficiently wide range to cover present practice in compound locomotives:

	Pressure at *d*.	Mean press. bet. *d* and *e*.	Pressure at *e*.	Mean press. bet. *e* and *f*.	Pressure at *f*.	Mean press. in receiver.
$C = v$, $R = 2$.....	80.	91.8	106 7	91.8	80	91.8
$C = 1.5\,v$, $R = 2$.....	80.	88.9	100.	88.9	80	88.9
$C = 2\,v$, $R = 2$.....	80.	87.4	96.	87.4	80	87.4
$C = v$, $R = 2.5$...	72.	82.6	96.	77.8	64	80.2
$C = 2\,v$, $R = 2.5$...	69.3	75.7	83.2	72.4	64	74.

An examination of this table shows that, with the assumed cut-offs and relative volumes, the actual pressure in the receiver during one stroke may vary as much as 26.7 pounds, and that in all of the cases given the pressure at f, or the point of l. p. cut-off, is considerably below that at e, while the mean pressure between e and f does not differ much from the mean pressure in the receiver. In designing compound engines the pressure in the receiver is frequently assumed as constant.

It will be seen from the above that this assumption does not introduce any serious error as far as the h. p. back pressure and the l. p. mean pressure up to the point of cut-off are concerned. But as the pressure at f is considerably below the mean receiver pressure, the mean pressure between f and g, calculated on the basis of a constant receiver pressure, will be too high. In many cases, however, it is well-nigh impossible to predetermine the receiver pressure by calculations, and the best that can be done is to estimate it from the known pressure found in similar engines in practice. The nature of these difficulties will appear later, and for the present we will continue to calculate pressures, assuming as before that there is no condensation in the receiver. The table also shows clearly one result of changing the relative capacity of the receiver, viz., that the larger the receiver the smaller are the variations in pressure in it during a stroke.

FINAL PRESSURE; TOTAL EXPANSION.—In a compound engine, a certain fraction of the h.p. cylinder is filled with steam from the boiler at each stroke, and after expanding in both cylinders this mass of steam finally fills the l. p. cylinder before it is exhausted into the atmosphere or condenser. For example, if the cylinder ratio is 2.5 and the h. p. cut-off is at one-half stroke, $.5\,v$ is the volume admitted from the boiler at each stroke, and this finally fills the volume $2.5\,v$ before it is exhausted. The steam is therefore expanded to five times its initial volume, or the ratio of total expansion is 5, and the final pressure at which it is exhausted will be one-fifth of the initial pressure, or 32 pounds in the case we have used for purposes of illustration. Similarly, if the h. p. cut-off was at three-eighths stroke the ratio of total expansion would be $2.5 \times \frac{8}{3} = \frac{20}{3} = 6\frac{2}{3}$, and the final pressure in the l. p. cylinder would be three-twentieths of $160 = 24$ pounds. It will be noted that the only data required in determining the total expansion and final pressure are the ratio of the cylinders and the h. p. cut-off, or, in other words, these results are independent of the capacity of the receiver and of the l. p. cut-off. The effect of the size of the receiver is seen in the shape of the indicator cards due to the compressions and expansions; but how many or how large these variations are does not affect the final pressure. The office of the l. p. cut-off is to control the division of the work between the two cylinders. In a compound engine, which exhausts into the atmosphere, the steam can be expanded economically until the boiler pressure is reduced to the atmospheric pressure. Steam at 160 pounds absolute could, therefore, be expanded $\frac{160}{14.7} = 11$ times, nearly, if it were not for losses of pressure by wire-drawing, condensation, etc. In two-cylinder compound locomotives, large ratios of expansion are practically unattainable on account of the large l. p. cylinders necessary.

Drop in Pressure in Receiver.—We are now prepared to show how the receiver pressures which have been assumed can be calculated. Taking as before, $R = 2.5$, $C = v$, h. p. cut-off at one-half stroke, and l. p. cut-off at one-half stroke, we have the final pressure at the end of the expansion in the l. p. cylinder equal to one-fifth of 160, or 32 pounds. The ratio of expansion in the l. p. cylinder is 2, therefore the pressure at the point f is $32 \times 2 = 64$ pounds. Then, knowing the ratio of expansion from e to f, as already calculated to be 1.5, we have the pressure at $e = 64 \times 1.5 = 96$ pounds, which was assumed for the time in calculating the variations of pressure in the receiver. Working back from this still further, we find the pressure at d as before, 72 pounds, and as the pressure at c is 80 pounds, there has been a *drop* in pressure of 8 pounds when the h. p. exhaust opened. When the l. p. steam valve closed at f, the pressure of the steam left in the receiver was 64 pounds. Then when the h. p. exhaust opened, the steam which filled the h. p. cylinder at a pressure of 80 pounds mixed with this, and gave a resulting pressure of 72 pounds. This drop represents an actual loss in the efficiency of the steam in the engine, since when it occurs the steam expands without doing useful work. It can be readily avoided, but in remedying this defect we may introduce others which are of more importance.

To prevent drop, it is only necessary to adjust the cut-off of the l. p. cylinder so that the volume of steam drawn by it from the receiver equals that of the h. p. cylinder. For instance, with the dimensions already given in this paragraph, it will be evident that when the l. p. cut-off is at $\frac{1}{2.5}$ or two-fifths of the stroke, there will be no drop, because two-fifths of the l. p. cylinder is equal to the whole h. p. cylinder in volume, and if we withdraw from the receiver at each stroke a volume which is equal to that received from the h. p. cylinder, the pressure in the receiver will not be reduced. This can also be readily shown by calculating back from the final pressure in the l. p. cylinder as before. Suppose *e f* to represent two-fifths of the l. p. stroke instead of one-half, as shown in the figure, then the pressure at *f* would be $32 \times \frac{5}{2} = 80$ pounds, which would be the pressure in the receiver when the h. p. exhaust opened ; and as this is also the final pressure in the h. p. cylinder, there would be no drop. There is, of course, always more or less drop due to wire-drawing and friction in passages which cannot be prevented, and it must also be borne in mind that all of these calculations are based on the assumption that pressures vary inversely as the volumes.

MEAN EFFECTIVE PRESSURE ; EQUIVALENT PRESSURE IN ONE CYLINDER.—With the data already used the mean forward pressure in the h. p. cylinder was found to be 135.5 pounds. The mean receiver pressure, or h. p. back pressure, is 80.2 pounds, and thus the mean effective pressure in the h. p. cylinder is $135.5 - 80.2 = 55.3$ pounds. For the l. p. card, the mean pressure between *e* and *f* was found to be 77.8 and the pressure at *f* was 64 pounds. By formula (2) the mean pressure between *f* and *g* is $64 \times .693 = 44.4$ pounds. The mean forward pressure for the stroke is then $\frac{77.8 + 44.4}{2} = 61.1$ pounds, and assuming a back pressure of 18 pounds, or 3.3 above the atmospheric pressure, the l. p. mean effective pressure will be 61.1 — 18 = 43.1 pounds.

As the ratio between the cylinder areas is 2.5, assuming the stroke to be the same in both cylinders, as it generally would be in practice, one pound per square inch on the l. p. piston is equivalent to 2.5 pounds per square inch on the h. p. piston. We can thus readily find the effective pressure in a single cylinder, which is equivalent to the effective pressure in the two cylinders of the compound engine. Ordinarily the mean pressure is thus referred to the l. p. piston, although a reference to the h. p. piston is more convenient for some purposes. In the present case, the effective h. p. pressure referred to the l. p. piston is $\frac{55.3}{2.5} = 22.1$. The total effective pressure referred to the l. p. piston is then 22.1 + 43.1 = 65.2 pounds. From this we find that the proportion of the total work which is done by each cylinder is, in h. p., $\frac{22.1}{65.2} = .34$, and in l. p. $\frac{43.1}{65.2} = .66$. If the pressures are referred to the h. p. piston, we have 43.1 × 2.5 + 55.3 = 107.8 + 55.3 = 163.1 as the equivalent pressure in one cylinder of the same size as the h. p. cylinder. A common practice has been to make the h. p. cylinder of a compound locomotive of the same size as one cylinder of the simple engine which it is intended to replace. On this basis the theoretical compound engine under discussion would be developing the same work as the simple engine when the latter was developing a mean effective pressure of one-half of 163.1 = 81.6 pounds in each cylinder.

EFFECT OF CHANGING THE POINT OF CUT-OFF.—If in Fig. 1 the h. p. cut-off is made earlier, while the l. p. cut-off remains as before at one-half stroke, a series of changes will be introduced, which are shown in full lines in Fig. 2, the lines of Fig. 1 being repeated in dotted lines. Assuming a cut-off at three-eighths stroke, the final pressure in the h. p. cylinder is 160 × $\frac{3}{8}$ = 60 pounds, or at *c′* instead of *c*. Also, as the total expansion is now $2.5 \times \frac{8}{3} = \frac{20}{3} = 6\frac{2}{3}$ instead of 5, the final pressure at *g* is reduced to *g'*, which represents $160 \times \frac{3}{20} = 24$ pounds. Then, as the h. p. cut-off is unchanged, the pressure at *f* is reduced to *f'*, or $24 \times 2 = 48$ pounds. The steam which fills the h. p. cylinder at a pressure of 60 pounds is mixed with an equal volume in the receiver at a pressure of 48 pounds, giving a resulting pressure at *d* of 54 pounds. The results of this change are, then, that the forward mean pressure in the h. p. cylinder, the pressure in the receiver, the initial pressure in the l. p. cylinder, and the mean pressure in that cylinder are all less than before. The work done by the l. p. cylinder is therefore less, while for the h. p. cylinder we have taken from one part of the card and added to another part. The total work done by both cylinders is, of course, less than before, but the proportion done by the h. p. cylinder is greater, and, in fact, the mean effective pressure in that cylinder has been increased. When both cut-offs were at the same point considerably more work was being done in the l. p. than in the h. p. cylinder, but by making the h. p.

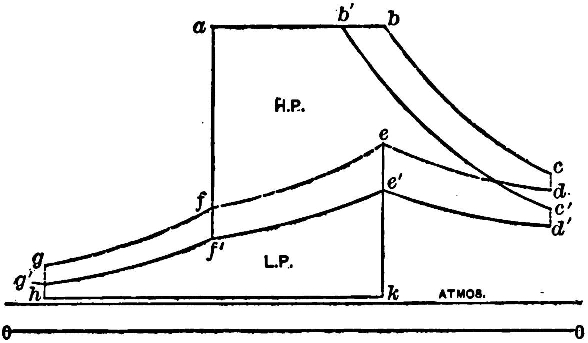

Fig. 2

cut-off the earlier of the two there is less difference in work than before, or, in other words, the work may be equalized by this means. A similar effect will, of course, be produced by making the l. p. cut-off *later* than that of the h. p., and conversely by making the l. p. cut-off earlier than that of the h. p. the proportion of the total work which is done by the l. p. cylinder will be increased. The following table calculated for $R = 2$ and $C = 1.5\ v$ will illustrate this:

Cut-off.		Mean press. h. p.	Mean press. l. p.	Mean h. p. press. referred to l. p.	Total mean in one cyl.	Propor'n of work.	
h. p.	l. p.					h. p.	l. p.
$\frac{1}{2}$	$\frac{1}{2}$	46.6	54.0	23.3	77.3	.3	.7
$\frac{3}{5}$	$\frac{1}{2}$	51 4	39.6	25.7	65 4	.4	.6
$\frac{2}{5}$	$\frac{3}{5}$	39.2	48.9	19.6	68.4	.29	.71
$\frac{1}{2}$	$\frac{3}{5}$	31.5	60.3	15.7	76.0	.21	.79

In practice the pressure in the receiver may be less than that calculated, on account of losses in the h. p. cylinder and passages. The effect of a lower receiver pressure is to increase the proportion of work done in the h. p. cylinder, so that by adjusting the valve gear to give an earlier cut-off in the h. p. cylinder than in the l. p., the total work may be very nearly equally divided between the two cylinders.

In Figs. 1 and 2 the l. p. cut-off has been taken at one-half stroke, and it was assumed that release occurred in the h. p. cylinder exactly at the end of the stroke. If now we make the l. p. cut-off later than one-half stroke, leaving everything else unchanged, there will be an exhaust from the h. p. cylinder, while the l. p. steam valve is still open, which will increase the pressure in the receiver and cause what may be called a re-admission in the l. p. cylinder. This is illustrated by Fig. 3, in which the h. p. exhaust occurs at *b*, causing a rise in pressure to *c*, from which there is expansion as before in the h. p. cylinder, the receiver and the l. p. cylinder until the l. p. steam valve closes at *d*. A similar effect will be produced by pre-release in the h. p. cylinder. An examination of a diagram such as Fig. 4 may make this subject more clear. In this Fig. *b c* represents the stroke of the pistons and the circle the path of the crank pins. Taking the direction of revolution as indicated by the arrow, when the h. p. piston is at the end of a stroke or its crank is at *a c*, the l. p. crank will be at *a c′*, and the exhaust from the h. p. cylinder which takes place at this position of the cranks will cause the rise in the l. p. card shown at *c*, Fig. 3. If the h. p. exhaust occurs before the end of the stroke, for example when the piston is at *d*, the l. p. crank will be at *a e′*, and the l. p. piston at *g*, causing a rise in the l. p. card as shown at *k*, Fig. 3. In cards taken from an engine this increase in pressure will of course be more gradual, and at high speeds may simply cause the l. p. admission line to be more nearly parallel with the atmospheric line.

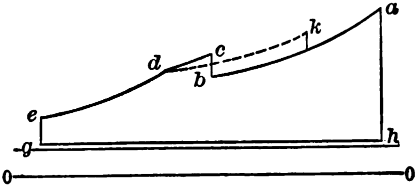

Fig. 3

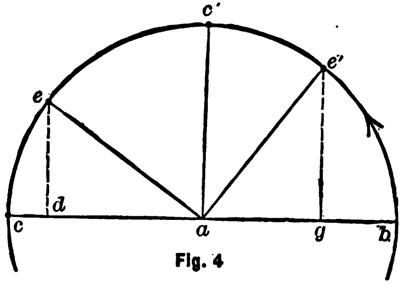

Fig. 4

CHAPTER II.

RATIO OF CYLINDERS.—In treatises on compound engines formulas are generally deduced for determining the ratio between the volumes of the two cylinders, so that the total work done by the engine will be equally divided between them. As usually given for receiver engines, these formulas are based either on the assumption that there is no drop in pressure in the receiver, or else it is assumed that the receiver pressure is constant. The rule most commonly given is that *R* equals the square root of the total number of expansions. In designing compound locomotives any such rule could be used only for a rough approximation at best, and in general would be of no value, since the requirements of construction place a maximum limit upon the size of the l. p. cylinder, which is less than that theoretically advisable.

The ratios which have been used for two-cylinder compound locomotives range from 2.74 for small engines to 1.77 for large engines. Mr. Mallet has stated that the ratio should not be less than two. Mr. von Borries, in his pamphlet on compound locomotives, recommends ratios of from 2 to 2.05 for large locomotives with tenders, and from 2.15 to 2.2 for tank locomotives.

These ratios have apparently been adopted by other designers, as we find that for the greater number of locomotives, of which records are published, the ratios lie between 2 and 2.1. With such ratios the division of the work between the cylinders is regulated by adjustments of the valve gear. Larger ratios than these are used for small two-cylinder locomotives, but it does not appear that ratios smaller than 1.9 have been used except in converted inside cylinder engines.

CLEARANCE.—In discussing the distribution and action of steam in a cylinder, the term clearance space, or simply clearance, is understood to mean the volume included between the piston when at the end of a stroke and the valve face at that end, and thus includes the steam port, the space between the piston and the cylinder-head, and any other spaces which are in communication with the foregoing, such as indicator pipes and cylinder drains. One of the principal effects of clearance is to make the effective cut-off later than it would be without clearance, and this produces results which can be best illustrated by reference to a figure. Referring to Fig. 5, let *e d* represent the stroke of a piston, and assume a cut-off at one-half stroke and 10 per cent. clearance. Then *a b* is one-half of *e d*, and the apparent ratio of expansion is two. Lay off *e f* equal to one-tenth of *e d*, then *f e* or *a g* represents the clearance. The volume which is filled with steam when cut-off takes place is *g b*, and this expands until it fills the volume of *f d*. The actual ratio of expansion is therefore *f d* divided by *g b*, or in the present case $\frac{1 + .1}{.5 + .1} = \frac{1.1}{.6} = 1.83$ instead of 2. Expressing this as a formula, the actual ratio of expansion $= \frac{1 + k}{n + k}$, in which k is the clearance expressed as a decimal of the volume displaced by the piston in one stroke, and n is the apparent cut-off, or one divided by the apparent ratio of expansion. The point c on the expansion curve is, of course, higher with a ratio of expansion of 1.83 than with a ratio of 2, and hence the mean pressure between b and c is higher. In making calculations the actual ratio of expansion should of course be used, but formula (1) will not then give correct results, as by it the mean pressure between g and c is found, and not that between a and c, and a correction must therefore be made which necessitates additional calculation. It is somewhat better to use formula (2), taking the corrected value for r, and still better in most cases to make use of a graphical construction. As an example of the application of formula (2), let the apparent cut-off be at one-third stroke with eight per cent. clearance. The actual ratio of expansion is then $\frac{1 + .08}{.33 + .08} = 2.63$, and the mean pressure between b and c will be $p_1 \frac{.967}{1.63} = .594\, p_1$. This is for two-thirds of the stroke, and for the first third the mean pressure equals p_1. The mean for the stroke is therefore $\frac{2 \times .594\, p_1 + p_1}{3} = .73\, p_1$. The mean pressure calculated by formula (1) without correction would be $.70\, p_1$.

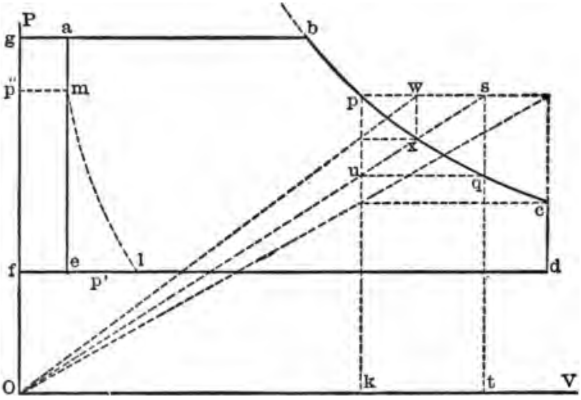

Fig. 5

CONSTRUCTION OF THE EXPANSION CURVE.—A simple method of plotting points on the expansion curve is the following, which requires only a triangle and a straight edge. In Fig. 5 let $O\ V$ be the zero line of pressures, $O\ P$ the zero line of volumes, and p a known point on the hyperbola. Through p draw $p\ s$ parallel to $O\ V$, making it of any convenient length. Draw *p k* and *s t* perpendicular to *O V* and draw *O s*. Through the point *u* where *O s* crosses *p k*, draw *u q* parallel to *O V*, and where this line cuts *s t* at *q* is a second point on the curve. Any number of other points can be found from *p* or *q* in a similar manner, as indicated in Fig. 5. An advantage of this method is that the distance of a point from *O P* can be selected at pleasure, as it will be always directly under the point to which the diagonal is drawn, as *q* and *s*, or *x* and *w*.

COMPRESSION.—In considering compression or cushion in compound locomotives, we find that it is a factor of the steam distribution which it is more difficult to dispose of satisfactorily than in simple engines. For economy of steam, the pressure in the clearance space when the steam valve opens should be equal to the initial pressure, while the necessary pressure for cushioning the reciprocating parts is a problem in itself.

There is, of course, no advantage in compressing to a pressure higher than the initial pressure. In a simple engine having an initial pressure of 175 pounds absolute, and a back pressure of 18 pounds absolute, it is possible to compress to 9.7 times the back pressure before the initial pressure will be exceeded. If the same pressures are taken in a compound engine, for the h. p. initial pressure and the l. p. back pressure, we have the same possible range of compression, but it is divided between two cylinders. If the receiver pressure is 70 pounds absolute, the possible range of compression is for the h. p. cylinder from 70 to 175 pounds, and for the l. p. cylinder from 18 to 70 pounds, or 2.5 times in the former and about 3.9 times in the latter. It will be at once apparent that the valve adjustment for compression in the compound is a much more difficult problem than in the simple engine. For example, with 5 per cent. clearance, and the pressures as just stated, the pressure in the clearance space at the end of the stroke would equal the initial pressure in the h. p.

cylinder when the exhaust closed at $2.5 \times .05 - .05 = .075$ of the stroke from the end, or at 92.5 per cent. of the stroke, as it is frequently stated. In the l. p. cylinder, an exhaust closure at 85.5 per cent. would fill the clearance space with steam at receiver pressure. With 10 per cent. clearance, and the same pressures as before, the earliest allowable points of exhaust closure would be 85 per cent. in the h. p. and 71 per cent. in the l. p. cylinder. It will be seen from this that a large percentage of clearance in a compound engine may be a positive advantage. The relations between the back pressure, the pressure from compression, the point of exhaust closure and the clearance, can be expressed in a general formula as follows: Referring to Fig. 5, let p' represent the back pressure and p'' the pressure in the clearance space at the end of the compression, both measured from the zero line of pressures; let l be the point of exhaust closure, $l\,m$ the compression curve which is considered as a rectangular hyperbola, $d\,e$ the stroke of the piston, and $f\,e$ equal k, the clearance as before. Then the fraction of the stroke at which the exhaust should close to produce a pressure p'' is

$$\frac{d\,l}{d\,e} = 1 - \left(\frac{p''}{p'} - 1\right)k. \quad (3)$$

The problem of determining the amount of compression necessary to cushion the reciprocating parts does not differ essentially in compound and simple engines, and for that reason will not be discussed at length at this time. The work done in compressing the steam from l to e, Fig. 5, exceeds that done by the steam on the other side of the piston during the same time, and this excess of work tends to retard the piston. It will be evident that in determining the amount of this excess we can regard the back pressure line as the zero line of pressures, if it is parallel to the atmospheric line, and then determine how much the area $l\,m\,e$ exceeds the area of an equal length from the other end of the card. The actual pressure of the back pressure line, or whatever line is used for a zero line from which to measure pressures, is therefore of no importance in this connection, and hence the problem will be very similar in all steam cylinders.

INDICATOR CARDS IN PRACTICE.—The indicator cards taken from compound locomotives in practice will, of course, differ greatly from those which have been called elementary theoretical cards, and which were illustrated by Figs. 1, 2 and 3. The cards taken from the engine should agree very closely with theoretical cards which are drawn according to a complete and correct theory. What has been called the elementary theory is not strictly correct, and is incomplete; but it is preferable for practical purposes to a more accurate construction on account of its simplicity, and because we cannot predict the exact form of an indicator card from an engine, which differs from existing types to an appreciable extent.

The causes which produce modifications of the elementary theoretical card, are chiefly the initial condensation and re-evaporation during expansion; the size, shape and location of the steam passages and the receiver; pre-release and compression; wire-drawing due to gradual opening and closing of ports, and the effects of high piston speed.

Indicator cards from a compound locomotive, in which steam was cut off at about four-tenths of the stroke in both cylinders, are shown in full lines in Fig. 6. The clearance space is 10 per cent. of the piston displacement in the h. p. cylinder, and 7.5 per cent. in the l. p. cylinder. The volume of the receiver is one and one-half times the h. p. cylinder. With this data the theoretical lines shown dotted in the figure have been constructed, making allowance for the excessive drop shown between the two cards. The differences between the actual admission and expansion lines of the h. p. card are the same as in cards from simple engines, and are due to the wire-drawing during admission and at cut-off, and to the re-evaporation during expansion.

The extent of these departures from the assumed theoretical curve varies greatly in simple engines, and probably depends upon the piston speed, together with apparently small differences in valve gear and steam passages. The only satisfactory way of determining the probable loss in a proposed engine, whether simple or compound, is to examine

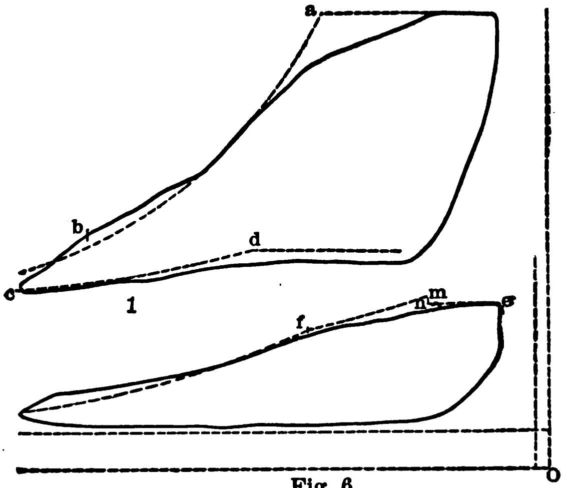

Fig. 6

indicator cards from an existing engine of the same general proportions, and having a valve gear of the same type and dimensions. This is also true of the loss of pressure between the boiler and the cylinder. Indicator cards taken from engines of various makes when on similar service show variations in these particulars of as much as 20 per cent., and it is obvious that no general rule can be laid down.

In Fig. 6, when the h. p. exhaust occurs at *b*, the l. p. piston is at *n*, and re-admission to the l. p. cylinder takes place, causing a rise in pressure to *m*. The l. p. piston moves from this position to that of cut-off *f*, four-tenths of the stroke, before the h. p. piston has moved over the remainder of its stroke from *b* to *c*. The pressure at *c* was calculated approximately on the basis of the receiver pressure when the h. p. exhaust opened, being that at *f*; but if the valve opened with sufficient rapidity this point would be found directly below *b*. From *c* to *d* there is compression as shown, and from *d* there is expansion until

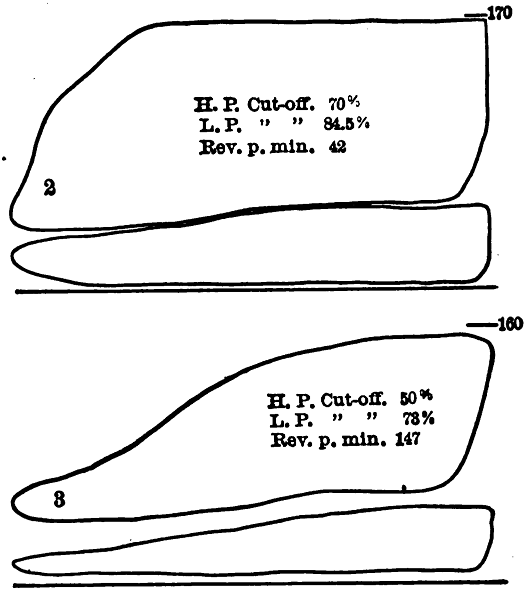

Fig. 7

compression in the h. p. cylinder begins; but with the proportions above given, this expansion line is nearly parallel to the atmospheric line, or in other words, there is practically no expansion. Turning now to the l. p. card, and taking the pressure at *e* as that of the steam in the receiver, we find that the line from *e* to *n* is practically at constant pressure, and that the rise in pressure from *n* to *m* is comparatively slight. Also, that during the expansion from *m* to *f* the fall in pressure is not great. The drop between the h. p. and the l. p. cards in this figure has been referred to as excessive, and data which would furnish an

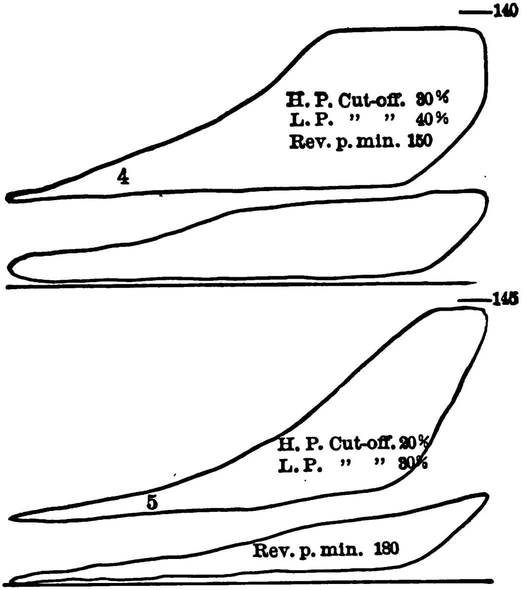

Fig. 8

explanation are not at hand, but this does not lessen the value of the cards for our present purpose.

In Figs. 7 and 8 are shown indicator cards from two-cylinder compound locomotives at different speeds and points of cut-off. The shape of the h. p. back-pressure line is to be noted. Cards Nos. 2 and 3 are from the same engine, and it will be noticed that the compression up to about the middle of the back stroke is quite marked, and that the remainder of the back-pressure line is nearly horizontal, as it was found in Fig. 6. In Nos. 4 and 5 the compression appears to continue during the whole of the back stroke. This is found to be the case in a considerable number of cards which have been examined, and is particularly noticeable at high speeds.

It will also be seen that the l. p. cards do not differ much in appearance from cards taken from simple engines. This would seem to indicate that we can without great error consider the receiver pressure as constant, and therefore treat the l. p. cards as if they were actually taken from simple engines. This arises largely from the effects of re-admission. It is not likely that, with the ordinary valve gear, the h. p. release will occur later than at 90 per cent. of the stroke, and an examination of a diagram of the crank positions such as Fig. 4, will show that the l. p. piston has moved over 20 per cent. of its stroke at this position.

The l. p. cut-off will not generally be earlier than three-tenths of the stroke, and hence it is safe to say that re-admission will always occur in practice. We have already seen that the practical effect of this is to make the l. p. admission line more nearly parallel with the atmospheric line, or, in other words, causes the l. p. admission line to more nearly resemble the admission line of a card from a simple engine.

In Fig. 9 are shown the admission and expansion lines of four indicator cards from the l. p. cylinder of a compound locomotive. The points of cut-off given are those which were recorded on the cards. The dotted lines indicate the form of the theoretical card for these points of cut-off and for the initial pressures as shown.

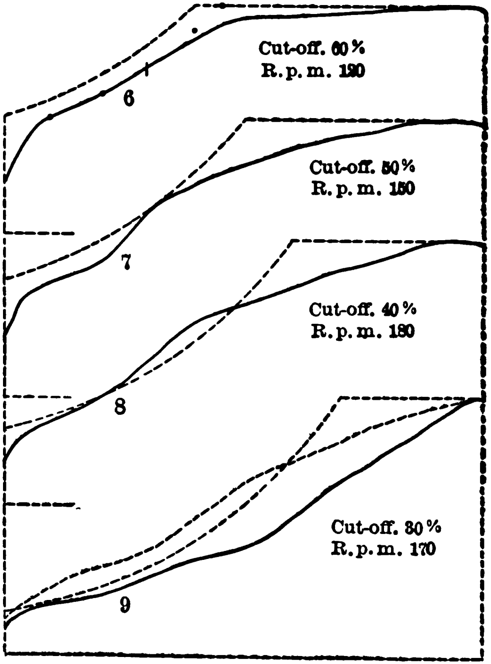

Fig. 9

On card No. 6 a curve which agrees with the actual curve very closely is indicated by dots, and shows an earlier cut-off than that recorded. On card No. 9 the irregular dotted line shows the form of the card from the other end of the cylinder with the same nominal point of cut-off. It will be apparent from Fig. 9 that the l. p. cards do not differ more from the elementary theoretical cards for simple engines than do the actual cards taken from these engines.

To obtain a fair idea of the probable form of the cards from a compound locomotive but one factor now remains to be determined, and that is the probable pressure in the receiver. On this hinges the division of work between the cylinders, and there is, in fact, no general rule by which it can be calculated with any certainty. The size of steam passages, and the size and location of the receiver, all steam space between the valves of the two cylinders being included in this term, will control this pressure to a great extent, as upon these depend the losses of pressure due to friction in passages and to condensation. Mr. von Borries states, in his pamphlet on Compound Locomotives, that for cut-offs of from 30 to 40 per cent. the pressure in the receiver should be from 30 to 33 per cent. of that in the boiler. If the pressure maintained in the receiver of an engine in practice is known, the probable receiver pressure in a similar proposed engine can of course be predicted with some degree of certainty; but when any different arrangement of valves and passages is used, the distribution in previous engines can serve as a guide only. The following is suggested as a method of approximating to the probable receiver pressures. With the same initial pressure in the h. p. cylinder, as the h. p. cut-off is made later the receiver pressure will become higher, and a similar effect is produced by making the l. p. cut-off earlier. Hence a formula for calculating the receiver pressure should have the form $p = c \times p_1 \frac{\text{h. p. cut-off.}}{\text{l. p. cut-off.}}$ In this formula p is the absolute receiver pressure, p_1 the absolute h. p. initial pressure, and c is a numerical coefficient.

An examination of a considerable number of indicator cards from compound locomotives gave an average value for c of 0.46, but this value is not recommended except for approximations, and of course no such formula can take the place of direct experiment.

COMBINED INDICATOR CARDS.—It is a common practice to combine the indicator cards taken from the several cylinders of compound engines in one diagram, and to compare this with an assumed curve of reference. The expansion of the steam in two or more cylinders can thus be compared approximately with an equal expansion in one cylinder, but it is not clear that much can be learned from such a diagram. The reference curve is ordinarily the rectangular hyperbola. Fig. 10 illustrates a combined diagram from a compound locomotive, of which the separate cards as taken closely resembled card No. 4 (Fig. 8). In Fig. 10 vertical distances represent pressures, and horizontal distances rep-

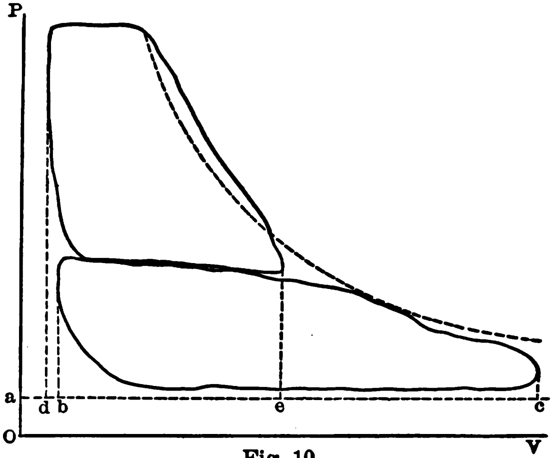

Fig. 10

resent volumes as usual. Both cards must first be reduced to the same scale of pressures. Then take any convenient distance such as *b c* to represent the volume of the l. p. cylinder, and let *a b* represent the volume of its clearance space. Then *O a P* is the zero line from which to measure volumes, and *O V* drawn as usual is the zero line of pressures. Lay off *a d* equal to the h. p. clearance space, and *d e* equal to the volume of the h. p. cylinder, both on the same scale as that of the l. p. cylinder; or *d e* should equal *b c* divided by the ratio of the cylinders. The outlines of the cards are then found by plotting points as usual.

CHAPTER III.

General Arrangement of Parts.—While the disposition of cylinders and steam chests with regard to the boiler and running gear in two-cylinder compound locomotives does not differ from European practice in simple locomotives, the same diversity of design, which has heretofore been remarkable in that as compared with American practice, is found in their compound locomotives. The designer will thus find precedent in existing engines for almost any arrangement of the principal parts, and for any type of valve gear which he is likely to adopt.

The credit of having inaugurated the present era of compound locomotives in Europe is due to Mr. Anatole Mallet, who designed successful two-cylinder compound locomotives for the Bayonne & Biarritz Railroad in 1876, and has since brought out many different designs. While it would not be incorrect to class the greater number of compound locomotives as belonging to the Mallet system, this term as applied to two-cylinder engines is usually restricted to those which can be operated either as simple or compound engines at the will of the engineer, as distinguished from others which are necessarily worked as compound engines, except for a brief interval in starting.

Engines of the latter class have been built in considerable numbers since 1880, especially after the designs of Mr. A. von Borries, Locomotive Superintendent of the Hanover Railroad, and Mr. T. W. Worsdell, Locomotive Superintendent of the North-Eastern Railway, England.

This class of engines comprise what is known as the Worsdell and von Borries system, the essential difference between their designs being in the method of accomplishing the automatic change from simple to compound working.

The von Borries System.—Figs. 11 and 12 illustrate the arrangement of cylinders and steam connections in two designs of compound locomotives according to the von Borries system. In both figures *h* is the h. p. cylinder, *l* is the l. p. cylinder, *A* is the steam pipe from the boiler to the h. p. cylinder, *C* is the receiver connecting the two cylinders, *V* is the starting and intercepting valve, *B* is the auxiliary steam pipe from the boiler to the starting valve, and *D* is the exhaust pipe from the l. p. cylinder. It will be noted that in Fig. 12 the steam pipes, receiver and exhaust pipe are almost entirely inclosed in the smoke-box, and it is very desirable that they should be so placed.

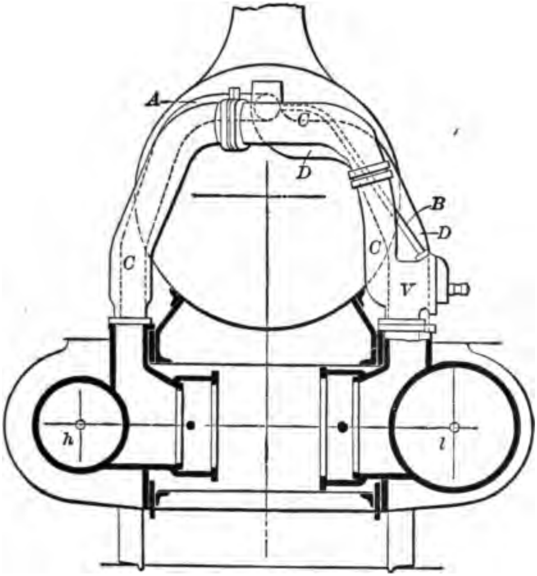

Fig. 11

The arrangement in Fig. 11 is that of a six-coupled freight engine for the Prussian State Railways, which was illustrated in *Engineering* of Feb. 1, 1889. The dimensions, which are of immediate interest in this connection, are as follows:

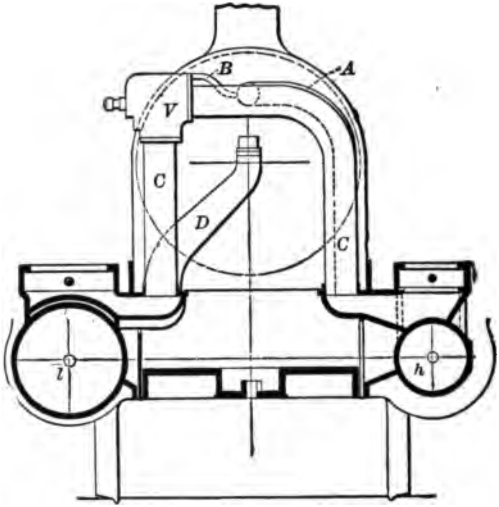

Fig. 12

Diameter of high-pressure cylinder	18.1 in.
" low-pressure "	25.6 "
Stroke of pistons	24.8 "
Diameter of driving wheels	52.4 "
Boiler pressure, gauge	174. lbs.
Diameter of boiler, smallest inside	50.8 in.
Tubes, 184, 2-in., length	14 ft. 7 in.
Grate area	16 sq. ft.
Heating surface	1,420 "
Weight in working order, all on driving wheels	88,250 lbs.
Valve gear	Allan.

Fig. 12 is taken from a paper on Compound Locomotives by Mr. R. Herbert Lapage, read in November, 1888, before the Institution of Mechanical Engineers (England), and illustrates a four-coupled passenger locomotive of the Santa Fé & Cordoba Great Southern Railway. The principal dimensions are as follows:

Diameter of high-pressure cylinder	16 in.
" low-pressure "	23 "
Stroke of pistons	24 "
Diameter of driving wheels	66 "
Boiler pressure, gauge	170 lbs.
Weight in working order	86,200 "
Valve gear	Stephenson.

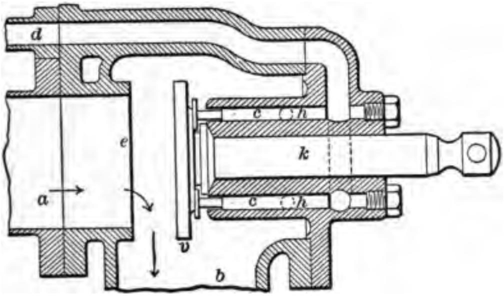

Fig. 13

The essential feature of the von Borries system is the combined intercepting and starting valve, a recent form of which is illustrated by Fig. 13. In this figure *a* is the receiver pipe which leads from the h. p. cylinder and *b* is the passage to the l. p. cylinder. The valve is shown in the position which it occupies ordinarily, or when the locomotive is working as a compound engine, the direction of the flow of the steam being as indicated by the arrows. Connected to the back of the intercepting valve *v* are two small plungers *c c* which together form the starting valve. Supposing the valves to be in the positions shown in Fig. 13 and the engine about to start, when the throttle is opened steam will be admitted to the h. p. cylinder by the usual pipe, and also to the auxiliary steam pipe *d*, and by the passage shown, to the back of the plungers. The pressure on the ends of the plungers is sufficient to move the intercepting valve *v* to the left in the figure until it seats at *e*. By the same movement two small ports *h h* are uncovered, through which steam from the boiler is admitted to the passage *b* and thence direct to the l. p. steam chest, while, as the intercepting valve is closed, this pressure does not act against the h. p. piston.

As the engine starts and the exhaust from the h. p. cylinder takes place, the pressure in the receiver rises until it is sufficient to overcome the pressure on the l. p. side of the intercepting valve, when this valve is moved back to the position shown in the figure, while at the same time the two small steam ports are closed by the plungers, and the engine begins to work as a compound. It is said that in practice the pressure of the steam from the boiler which is admitted to the l. p. cylinder is reduced by wire-drawing, due to the small steam pipe and ports, to about one-half the boiler pressure, and as the ratio of the cylinders is about two, the total pressure on the two pistons in starting is nearly equal. To prevent excessive pressure in the l. p. cylinder and receiver a safety valve is placed on the latter.

The pressure in the receiver when running is sufficient to overcome the boiler pressure acting on the ends of the two small plungers, together with the atmospheric pressure on the stem of the large valve *v*, and, therefore, the valves are maintained in the position shown in Fig. 13 as long as the throttle is kept open. In the paper referred to above, Mr. Lapage states that: "In practice it takes only from half to one revolution before the exhaust from the high-pressure cylinder opens the intercepting valve and closes the starting valves."

It will be noted that according to this statement, an engine with 66-inch driving wheels would begin to work as a compound after moving through from about 8½ to 17 feet. Just how far it will move before opening the intercepting valve will depend upon the positions of the cranks before starting, as will be demonstrated when considering the subject of the starting power. Strictly speaking, these locomotives can never work as simple engines, since the h. p. cylinder always exhausts into a closed receiver and never directly into the atmosphere. Mr. Lapage also stated

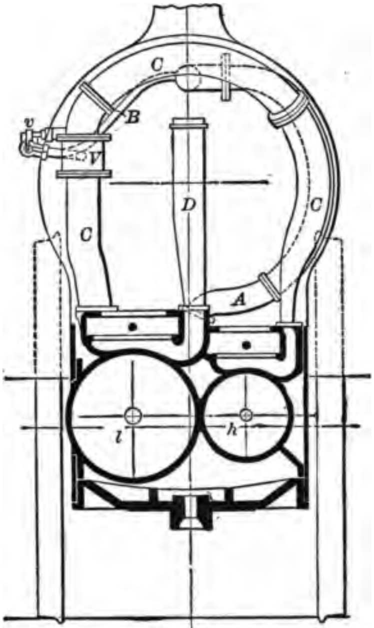

Fig. 14

that this automatic starting and intercepting valve had been already applied to about one hundred and fifty engines.

The earlier forms of intercepting valves were not wholly automatic in their action, but required to be closed by hand before opening the throttle in starting. In this form there were no small plungers, and the steam was admitted around the valve stem *k*, which was fluted for part of its length for this purpose. The valve was also connected by a bell-crank arrangement to a weighted arm, which held the valve open and prevented rattling when running with steam shut off.

THE WORSDELL SYSTEM.—Fig. 14 illustrates the arrangement of the cylinders and steam connections of a Worsdell four-coupled compound locomotive for passenger service, drawings for which were published in *Engineering* of March 30, 1888. More recent engines do not differ from this in regard to the cylinders and steam connections. In Fig. 14, *h* and *l* represent the high and low-pressure cylinders, respectively, *A* is the high-pressure steam pipe, *C* is the receiver, *D* is the low-pressure exhaust pipe, *B* is the steam supply to the starting valve *v*, and *V* is the intercepting valve.

The principal dimensions of this engine are as follows:

Diameter of high-pressure cylinder	18	in.
“ low-pressure “	26	“
Stroke of pistons	24	“
Diameter of driving wheels	80¼	“
Boiler pressure, gauge	170	lbs.
Diameter of boiler, smallest inside	53	in.
Tubes, 242, 1¾ inches, length	10 feet 11⅛	“
Grate area	17.33	sq. ft.
Heating surface	1,323.3	“
Weight in working order, total	97,000	lbs.
“ “ “ “ on driving wheels	68,000	“
Valve gear		Joy.

The Worsdell starting and intercepting valves are illustrated by Figs. 15 and 16, which are reproduced from *Engineering*. The intercepting valve is a flap valve, and is shown in Fig. 15 in the position which it occupies when the engine is working as a compound, being swung to one side, and thus leaving a straight, clear passage by it. The spindle on which the valve turns passes out through the side of the smoke-box, and carries an arm, which is connected to the small piston shown at *a*, Fig. 16, in a manner which is clearly indicated in the figures. The starting valve casing is connected to the main steam pipe by a small pipe, which is shown in Fig. 16 and also in Fig. 14. The piston

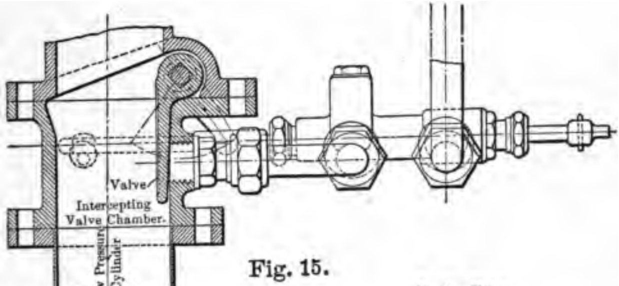

Fig. 15.

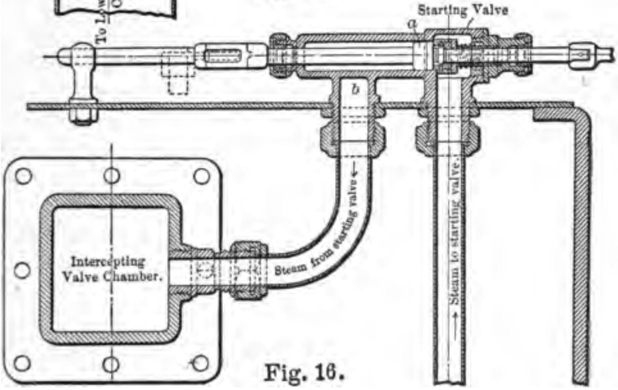

Fig. 16.

a, which operates the intercepting valve by means of the connection previously referred to, works in a cylinder which is an extension of the starting valve casing.

A small port, which is covered by a spring-loaded valve, connects this cylinder with the pipe *b*, and thus to the intercepting valve chamber. The starting valve is operated by a lever, and is a double valve, a slight movement of the lever opening the smaller valve, and further motion opening the larger valve, which is then partially balanced.

The operation of these valves in starting is as follows: The starting valve being opened by the engineer, steam, at boiler pressure, acts upon the small piston *a*, and moves it forward or to the left in the Figs. By the same movement the intercepting valve is swung up and closed, and the port connecting with the pipe *b* is uncovered, thus admitting steam from the boiler to the intercepting valve chamber below the valve, and thence to the low-pressure steam chest. As the exhaust takes place from the h. p. cylinder, the pressure in the receiver, above the intercepting valve, rises until it is sufficient to open that valve, when, by its movement, the small piston *a* is returned to the position shown in Fig. 16, and the steam supply is thus shut off. It will be seen that the Worsdell starting and intercepting valves are the same in principle as the von Borries valves, although the former are not wholly automatic in their action.

Valve Gear Adjustments.—In discussing the theory of two-cylinder compound engines, it was shown that the division of the total work between the cylinders could be equalized by maintaining a considerably earlier cut-off in the h. p. than in the l. p. cylinder.

Mr. von Borries finds that in practice for cylinder ratios of from 2 to 2.05 a cut-off of 0.4 in the h. p. cylinder calls for a cut-off of 0.5 in the l. p. cylinder. This adjustment gives a very nearly equal division of the work, and can be made without difficulty or increased complication with the valve gears which are suitable for locomotives, such as the Stephenson, Allan, Joy and Walschaert.

For the American form of the Stephenson shifting link motion, and when the locomotive is to run principally in forward gear, only a very simple change is necessary. It is to make the h. p. link hanger slightly shorter than the l. p. hanger, leaving all other parts as for a simple engine. This causes the h. p. link to be a little nearer the mid position than the l. p. link for all points in the forward gear, the result being to make the h. p. cut-off the earlier of the two as required. For the above cylinder ratios the h. p. hanger should be about one-twentieth of the lift of the link shorter than that of the l. p. gear.

This difference in length, of course, produces the opposite effect in the backward gear, the l. p. cut-off being then the earlier. For this reason the full-gear notch is the only one cut in the quadrant for the backward gear. Mr. von Borries gives the following as the approximate cut-offs which are obtained by this means:

	Forward.					Backward.
High-pressure cylinder......	75	50	40	30	20	78
Low-pressure cylinder.......	78	59	50	41	32	75

A more detailed tabulation of the distribution effected by this form of adjustment, as applied to the locomotive illustrated by Fig. 12, is appended. This table is taken from the paper by Mr. R. H. Lapage, to which reference has already been made.

For locomotives which are required to run equally well in either direction, the alteration described above is evidently not applicable, but a similar effect is produced by making the mid-gear travel of the l. p. valve less than that of the h. p. valve. This is, of course, readily accomplished by reducing the angular advance for the l. p. eccentrics, the outside lap of the l. p. valve being also made less. For cylinder ratios of from 2.15 to 2.2 Mr. von Borries recommends making the mid-gear travel of the l. p. valve 0.9 of that of the h. p. valve, and reducing the outside lap

STEAM DISTRIBUTION IN COMPOUND LOCOMOTIVE ON THE SANTA FE & CORDOBA GREAT SOUTHERN RAILWAY.

Angle of eccentrics, forward, 60½ *degrees; backward,* 61½ *degrees.*

HIGH-PRESSURE CYLINDER.

Sixteen inches diameter by 24 inches stroke. Clearance in link, top, 0.94 inches; bottom. 0.31 inches. Slip of block, top, 0 81 inches; bottom, 0.69 inches. Mid-gear travel, 2.63 inches. Outside lap, 1.12 inches; inside clearance, 0.25 inches. Length of ports, 12.87 inches. Length of link hanger, 14.06 inches.

	Notch in Quadrant.	Travel of Valve.	Lead.	Maximum Opening of Steam Port.	Cut-off, Per cent. of Stroke.	Release, Per cent. of Stroke.
		Inches.	*Inches.*	*Inches.*		
Forward gear	Full	4.19	0.02	1.00	73¾	89½
			0.02	0.94	72¼	88½
	5	3.50	0.06	0.63	61¼	83¼
			0.06	0 62	58½	81½
	4	3.19	0.09	0.47	52¼	78¾
			0.10	0.47	49	76¾
	3	2.97	0.12	0.34	41½	73½
			0.13	0.37	38½	70¾
	2	2.72	0.13	0.24	29½	66¾
			0.16	0 28	27½	63
	1	2.62	0.13	0.16	16½	58
			0.16	0 22	18	54
Backward gear.	For backward gear the full-gear notch is the only one cut in the quadrant.	2.72	0.12	0.25	26½	61¾
			0.13	0.22	20	59½
		2.91	0 10	0.38	37½	69¼
			0.12	0.28	31	68½
		3.09	0.09	0 47	47	75¼
			0.10	0.38	42½	75½
		3.37	0.06	0.62	55¾	79¾
			0.09	0.51	53¼	81
		3.62	0.03	0.74	62¾	83¾
			0.06	0.62	61¾	84½
	Full.	4.26	−0.03	1.07	74	89¼
			0.00	0.95	74½	90½

in about the same proportion. The approximate relative cut-offs thus obtained are as follows, being the same in both forward and backward gear:

High-pressure cylinder	75	60	50	40	30
Low-pressure cylinder	78	64	55	45	34

STEAM DISTRIBUTION IN COMPOUND LOCOMOTIVE ON THE SANTA FE & CORDOBA GREAT SOUTHERN RAILWAY.

Angle of eccentrics, forward, 60½ degrees; backward, 61½ degrees.

LOW-PRESSURE CYLINDER.

Twenty-three inches diameter, by 24 inches stroke. Clearance in link, top, 0.31 inches; bottom, 0.94 inches. Slip of block, top, 0.81 inches; bottom, 0.62 inches. Mid-gear travel, 2.56 inches. Outside lap, 1.12 inches; inside lap, 0.00. Length of ports, 17.00 inches. Length of link hanger, 14.64 inches.

	Notch in Quadrant.	Travel of Valve.	Lead.	Maximum Opening of Steam Port.	Cut-off, Per cent. of Stroke.	Release, Per cent. of Stroke.
		Inches.	*Inches.*	*Inches.*		
Forward gear.	Full.	4.50	−0.03	1.16	78	93¾
			−0.03	1.09	77½	94
	5	3.72	0.03	0.87	67¼	90
			0.06	0.87	66	89¾
	4	3.47	0.06	0.87	60¼	87¼
			0.09	0.87	57¾	86½
	3	3.22	0.09	0 84	51½	83½
			0.10	0.86	48½	82¾
	2	2.97	0.12	0.72	40	79¼
			0.13	0.74	37¾	78
	1	2.78	0.13	0.61	27¾	73¼
			0.16	0.66	27½	71
Backward gear.	....	2.62	0.13	0.55	17	62½
			0.16	0.56	13½	61½
	....	2.74	0.12	0 62	27¼	70
			0.16	0.61	22	70¼
	....	2.90	0.09	0.72	36¾	75¾
			0.16	0.69	31¾	77
	....	3.12	0.09	0.84	47½	81
			0.13	0.78	43¾	82½
	....	3.37	0.06	0.87	55½	85
			0.12	0.87	53½	86½
	Full.	3.90	−0.03	0.87	68½	90½
			0.03	0.87	69¼	91¾

Similar variations in the points of cut-off can be obtained with the other types of valve gears which have been mentioned, for example, by placing the arms on the lifting shaft of the Allan motion at an angle with each other, and by inclining the sliding links of the Joy gear. Other minor changes which will produce similar results will naturally occur to the designer.

For inside valve gear and outside steam chests Mr. von Borries advises the use of the American form of the Stephenson motion; for inside gear and inside steam chest, the Stephenson and Allan types; and for outside gear, the Walschaert motion. All of the recent Worsdell engines, concerning which information is at hand, are fitted with the Joy valve gear.

CHAPTER IV.

THE MALLET SYSTEM.—The Mallet system of two-cylinder compound locomotives has been already referred to as that in which by means of suitable valves the engine may be operated as a simple engine not only in starting, but at any time when in service. Such an engine, while having all the advantages of compound working, possesses an emergency power equal, or possibly superior, to a simple engine having the same general dimensions.

Figs. 17 to 20, inclusive, illustrate the arrangement of this system as applied to a converted six-coupled engine of the Western Switzerland Railroad. For the drawings from which the illustrations were made, and also for other data, I am indebted to Mr. Mallet. The cylinders are 17.7 and 25.6 inches in diameter by 25.6 inches stroke, one of the old cylinders having been retained as the h. p. cylinder. The driving wheels are 58.4 inches in diameter, and the working weight, which is all on the driving wheels, is 79,350 pounds.

In Fig. 17, *h* and *l* are the h. p. and l. p. cylinders, respectively, *A* is the main steam pipe from the boiler to the h. p. cylinder, *B* is the receiver, *C* is the l. p. exhaust pipe, *D* is the starting valve which is connected to the boiler by the pipe *E*, *F* is the intercepting valve, and *G* is the exhaust pipe from the h. p. cylinder when working as a simple engine.

The construction of the starting valve is shown in Figs. 18 and 19. It consists primarily of a short slide valve *a*, which, as shown, covers two ports leading to the receiver. The pipe *p* connects the starting valve chamber with the main steam pipe. On the back of the valve *a* is an inverted slide valve *b*, which slides on a seat formed in the valve-chest cover. A small pipe *c* connects the starting valve chamber with the intercepting valve, on the other side of the smoke box, as shown at *c*, Fig. 20. Referring now to Fig. 20, it will be seen that the intercepting valve consists of two circular valves and a piston, all being mounted on one stem, and so forming a sort of balanced double poppet valve. The connections to the intercepting valve are as indicated in the figure, the central opening connecting with the h. p. exhaust, the left with the common exhaust nozzle and the right with the receiver pipe.

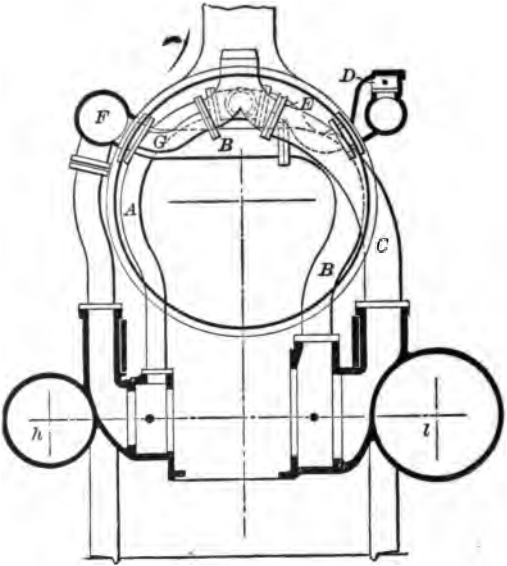

Fig. 17

The operation of these valves is as follows: They are shown in the illustrations in the positions which they ordinarily occupy, or when the engine is working as a compound. Under these circumstances steam from the boiler is admitted to the space *d* back of the piston *e* by way of the small pipe *c*, the starting valve chamber, and the pipe *p*. The pressure thus acting upon the piston *e* keeps the valve *g* closed against the ordinary receiver pressure. The intercepting valve can, of course, be connected so as to be worked by hand in connection with the starting valve. If now the starting valve is opened, or moved to the right in Fig. 18, steam from the boiler is thereby admitted to the receiver, and at the same time the pipe *c* is placed in communication with the atmosphere by means of the cavity in the top of the starting valve. The pressure back of the piston *e* being thus reduced, the valve *g* is opened by the receiver pressure, and the valve *h* is closed, in which position it is retained by the excess of the pressure in the receiver *k*, Fig. 20, or that on the l. p. side of the valve, over that on the h. p. side, which is now in communication with the exhaust nozzle. It will be seen that the locomotive will now work as a simple engine, and will continue to do so as long as the starting valve is kept open. As soon as it is closed, the intercepting valve will be returned to the position shown in Fig, 20.

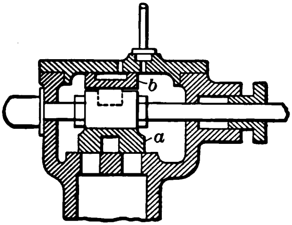

Fig. 18

On the engine illustrated by Fig. 17, a pressure-reducing valve is inserted between the starting valve and the receiver. This reducing valve is of the common differential piston type, adjusted by springs. In addition to this the receiver is fitted with a spring safety valve loaded to 70 pounds pressure. It would seem when a starting valve of this form is used in conjunction with a safety valve, that the introduction of a reducing valve is unnecessary, as the receiver pressure can be regulated by the starting valve.

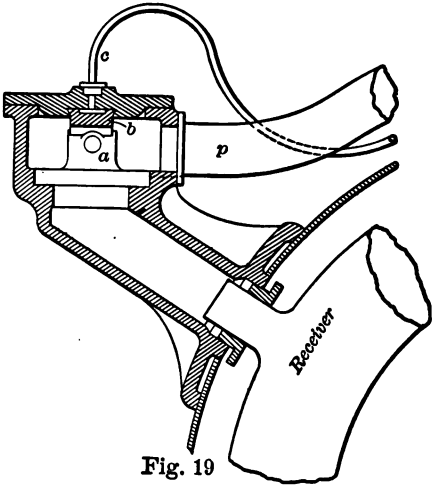

Fig. 19

In earlier designs Mr. Mallet has combined the starting and intercepting valve in one distributing valve. This is illustrated by Figs. 21 and 22, which are reproduced from *Engineering*. The distributing valve and a reducing valve are enclosed in a casing which is fastened to the smoke box. The main steam pipe is connected at *a*, and thence by a passage *b*, back of the valves, to the h. p. steam chest. An opening at *c* admits steam from this pipe to the reducing valve chamber and thence to the distributing valve chamber. The distributing valve is a slide valve, and covers three ports, as shown. Of these *d* is the h. p. exhaust, *e* connects with the receiver, and hence with the l. p. steam chest, and *g* leads to the exhaust nozzle. The valve is shown in the position for compound working. If it is moved forward, or to the left in the illustrations, the passage *d* is connected with *g*, and the h. p. cylinder exhausts directly to the exhaust nozzle, and at the same time by means of the passages *c* and *e* boiler steam at reduced pressure is admitted to the receiver and the l. p. steam chest.

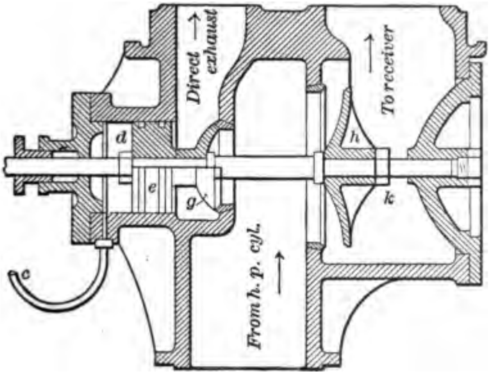

Fig. 20

In the earlier Mallet engines the lifting shaft is divided so that the valve motion of each cylinder is to a certain extent independent of the other. The h. p. valve gear is controlled by a screw and nut, which takes the place of the ordinary quadrant. The nut which is on the h. p. reverse lever carries a short sector or quadrant and a latch on the l. p. reverse lever works in this sector. The effect is that both cylinders can be reversed by moving the h. p. lever; while by adjusting

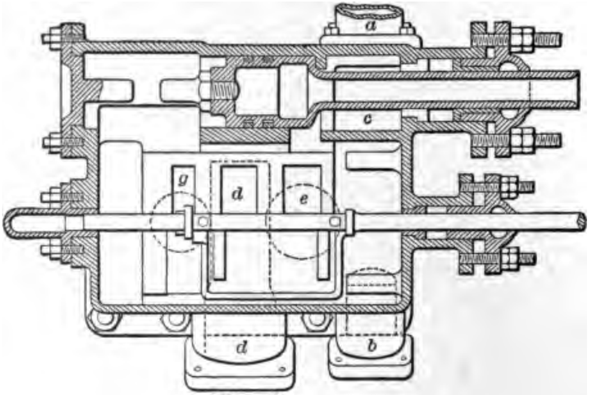

Fig. 21

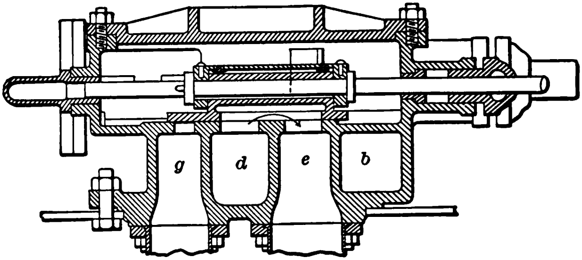

Fig. 22

the l. p. lever, the cut-off in that cylinder may be made either later or earlier than that in the h. p. cylinder. It is obvious that the same principle can be applied to the type of reversing gear common in this country by connecting the h. p. valve gear as at present, and connecting the l. p. reverse lever to the h. p. lever by means of an auxiliary quadrant carried by the latter. A device for tripping the latch of the l. p. lever might be advisable, so that the h. p. lever could be moved between the two extremes in both directions without regard to the position of the l. p. lever. It is, however, by no means proven that it is necessary or advisable to have two reverse levers, and, in fact, the weight of evidence appears to be to the contrary.

For the locomotive illustrated in Fig. 17, Mr. Mallet has adopted a differential motion for the purpose of obtaining a later cut-off in the l. p. cylinder in both forward and backward gear. The principle of this motion is illustrated by Fig. 23. In this Fig., *A* is the lifting shaft and *B* is an auxiliary shaft. The lifting arm *M* of the h. p. link and the arm *C* are keyed to the lifting shaft, while the l. p. lifting arm *N* and the arm *H* are in one piece, which turns about this shaft. The slotted arm *D* and the arm *E* are keyed to the auxiliary shaft. The arm *C* carries a block which slides in the slotted piece *D*. The parts are shown in the Fig. in a position for backing, the l. p. link being raised higher than the h. p. link and therefore cutting off later. In full backing gear the arms *M* and *N* would be parallel and hence give the same cut-off in both cylinders. In mid-gear the arms *C* and *D* are on the center line *A B*, while in forward gear, or to the left in the Fig., the lifting arm *N* is lowered more rapidly than the arm *M*. Mr. Mallet gives the following as the distribution obtained with this arrangement:

	Forward Gear.						Backward Gear.		
High-pressure cylinder...	.70	.60	.50	.40	.30	.0	.0	.60	.70
Low-pressure cylinder...	.70	.65	.60	.55	.50	.0	.0	.65	.70

STEAM PASSAGES.—It would seem to be an unquestionable principle of steam engine design, that all steam passages should be large and as free from abrupt bends and contractions as it is possible to make them, and should be so placed and protected that the loss by radiation will be reduced to a minimum. With these principles in view, there appears to be room for improvement in the designs of many existing compound locomotives in one or more particulars.

The proportional areas of steam ports in the two-cylinder compound locomotives which have been examined are somewhat less than in recent American locomotives. Mr. von Borries recommends areas which, for a cylinder ratio of two, would be about 8 to 8.2 per cent. of the piston area for the high-pressure, and 7 per cent. for the low-pressure cylinder for passenger engines. For freight engines having slower piston speeds, the areas may be 10 per cent. less. Mr. Worsdell's practice for both passenger and freight engines appears to be 8.1 per cent. for the high-pressure cylinder, and about 6.4 per cent. for the low-pressure. These proportions give for cylinders 18 and 26 inches diameter by 24 inches stroke; high-pressure steam ports, 1¾ by 11¾ inches; low-pressure steam ports, 2 by 17 inches; exhaust ports, 3½ inches wide and of the same length as the steam ports.

The volume of the receiver should not be less than that of the h. p. cylinder, and preferably should be larger than this, in order that the fluctuation in the h. p. back pressure may be as small as possible. Also in engines of the Worsdell and von Borries type a large receiver will permit a more satisfactory action in starting than a small one, as more time will be required to compress the exhaust steam from the high-pressure cylinder to the pressure necessary to open the intercepting valve. For engines of "ordinary size" Mr. von Borries gives 200 mm. (7⅞ inches) as a suitable diameter for the receiver pipe, with a thickness of about ¼ inch when made of copper.

Slide Valves.—The most noticeable difference between the distributing valves of compound and ordinary locomotives is in the large inside clearance of the former. This is made necessary by the high pressure of the back pressure line of the h. p. cylinder, and the consequent necessity of securing a late exhaust closure in order to avoid excessive compression in that cylinder. For the l. p. cylinder there are similar requirements, as, with the same back pressure as in simple locomotives, the pressure at the end of compression should not exceed that in the receiver. The necessary changes from the ordinary valves may perhaps be best illustrated by giving a few examples of the proportions which have been adopted:

	Width of port.	Travel.	Outside lap.	Inside clearance.	Lead.
H. P.	1¾	5¼	1⅛	¼	3/16
L. P.	2	5¼	15/16	⅛	3/16
H. P.	1¾	5¼	1⅛	¼	3/16
L. P.	2	5¼	1⅛	⅛	3/16
H. P.	1.6	3*	1.4	.3	.1
L. P.	2	3	1.4	0.0	.1
H. P.	1.35	2.65	1.3	.26	.1
L. P.	1.44	2.58	1.15	0.0	.06
H. P.	1	4.2	1.12	.25	.02
L. P.	1.16	4.5	1.12	0.0	— .03

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* Mid-gear travel. These dimensions are advised by Mr. von Borries.

These dimensions are given simply as illustrations, as of course the proper dimensions for any engine can only be determined satisfactorily by the use of a model, or by drawing the motion to full size. In either case adjustments by the indicator should follow, as the economy of the engine will depend very largely upon the care and attention given to the steam distribution.

SIZE OF CYLINDERS—TRACTIVE POWER.—A comparison of existing compound locomotives and of the rules suggested by different writers, shows about as much difference of opinion as to the proper proportions of cylinders, boiler pressure, driving wheel diameter and adhesion weight, as was found in ordinary locomotives by the committee of the Master Mechanics' Association on the subject in 1887.

The following is translated from the fourth edition (1888) of Mr. von Borries' pamphlet on Compound Locomotives: "To compute the size of the cylinders, we suppose, for the sake of simplicity, that the whole work is to be done in the large cylinder, the diameter of which is then computed by the following formula: $d^2 = \frac{2\,Z\,D}{p\,h}$. In which d = the diameter of the l. p. cylinder; Z = the required tractive force = 0.15 of the adhesion weight (where allowance is made in Z for the external engine friction, taken as equal to that for cars); D = the diameter of the driving wheels; p = mean effective pressure (after deducting internal machine friction); h = stroke of piston. The value of p depends upon the relative volumes of the two cylinders, and from experiments, and indicator cards may be taken from the following table:

	Ratios of cylinders.	p for 176 lbs. boiler pressure.
Large locomotives with tenders.	1:2 to 1:2.05	74 lbs. (42 per cent.)
Tank locomotives...............	1:2.15 to 1:2.2	71 lbs. (40 per cent.)

For locomotives for roads having many and long grades a larger co-efficient of adhesion ($Z = 0.16$) is to be used, to obtain larger cylinders; but 0.15 is ordinarily sufficient."

"For compound passenger and express locomotives, the diameter of the large cylinder may be made 1.5 that of the cylinders of the ordinary locomotive for the same service, the steam pressure being at the same time increased 15 to 30 pounds. This is based upon the supposition that the ordinary locomotive is properly proportioned. The working conditions should also be kept in view, so that the locomotives may work as much as possible at the more economical points of cut-off of 0.3 to 0.4 in the small cylinder."

For a passenger engine of 57,300 pounds on the driving wheels, Mr. von Borries gives the cylinder dimensions as 16.5 and 23.6 inches in diameter by 22.8 inches stroke, and says that such an engine will accomplish 10 to 15 per cent.

more than a common locomotive having about the same heating surface.

The following statement is made in the paper by Mr. Lapage, from which I have already quoted: "It is customary, when not otherwise specified, to make the size of the high-pressure cylinder the same as that of the cylinder of an ordinary locomotive. Such engines are found to haul from 5 to 10 per cent. heavier trains, while saving about 10 per cent. of fuel."

In a paper on "The Compound Principle as Applied to Locomotives," read by Mr. Edgar Worthington in January, 1889, before the Institution of Civil Engineers, we find it stated that the high-pressure cylinder should be made about one inch larger in diameter than the cylinders of an ordinary engine of equal power, even though an advance of 30 pounds per square inch of boiler pressure was contemplated for the compound engine.

The difference of opinion among designers may perhaps be best illustrated by a short table.

Weight on driving wheels. Pounds.	Diam. of driving wheels. Inches.	Boiler Press're. Pounds.	Stroke. Inches.	Actual diam. of cylinders. Inches.		Diameter by	
				h. p.	l. p.	Von Borries' Formula. l. p.	Mast. Mech. Formula, Simple.
57,320	73.2	176	22.8	17.3	24.8	27.3	17.5
57,300	68.1	176	22.8	16.5	23.6	26.4	16.9
88,250	52.4	174	24.8	18.1	25.6	27.6	17.2
68,000	80.25	170	24.0	18.0	26.0	30.9	19.8
40,320	85.25	175	24 0	18.0	26.0	24.2	15.5
93,700	61.25	160	24.0	18.0	26.0	32.0	20.3
101,600	61.25	160	24.0	18.0	26.0	34.0	21.8
46,480	42.5	160	21.6	10.6	17.0	20.2	13 0
....	60.00	120	24.0	18.0	24.0	....	*16.0
79,350	58.4	140	25.6	17.7	25.6	30.4	19.5

* Original diameter, simple engine.

In the last column of the table are given the diameters of cylinders for simple locomotives having the same boiler pressure, adhesion weight, and diameter of drivers, calculated by the Master Mechanics' formula. In the column preceding this are given the diameters of the low-pressure cylinders calculated by the von Borries formula, as already quoted. It will be seen that this formula gives, in general, larger results than are found in practice. The same formula, as quoted in the 1886 edition of "Recent Locomotives," contains values for Z and p, which will give diameters about 7 per cent. smaller than with those quoted above, which indicates that Mr. von Borries has concluded that larger cylinders are advisable.

It appears to the writer that no general rule can be devised which would be applicable to all cases, and that the character of the work which is to be required of the compound engine, as shown by a study of the work done by the simple engine which it is intended to replace, is the only safe guide. The maximum and minimum limits of mean pressure between which the engine will probably work, the usual demand upon it as determined by the character of service, and the type of starting device which is to be employed, must all be given consideration. In brief, if full advantage is to be taken of the economical possibilities of the compound locomotive, the first requisite in attempting its design is a thorough understanding of what it is expected to do.

The most satisfactory method of comparing the work done by simple and compound locomotives is a comparison of indicator cards taken from the two forms of engines under the usual working conditions. Complete sets of such cards from compound locomotives have not been obtainable, but a sufficient number of cards are included in the accompanying table to make it of some value. The figures given for the compound engines were selected principally from reports of the performance of various two-cylinder engines which have appeared from time to time in foreign technical journals, and may be assumed to be at least a not unfavorable representation of their capabilities. For the simple engines cards have been selected, which were taken at nearly the same number of revolutions as those of the compounds, in order to eliminate the question of speed as far as possible. For convenience in comparing the mean pressures in the two types of engines, the average of the h. p. mean effective pressure and that in the l. p. cylinder referred to the h. p. cylinder is given in column (a). It is, of course, not to be assumed that the

	COMPOUND.									SIMPLE.		
No.	Rev. per min.	Boiler pressure.	Initial press. h. p.	Cut-off, h. p.	Cut-off, l. p.	Mean press. h. p.	Mean press. l. p.	Mean l. p. press. ref. to h. p.	Average mean eff. pressure. (a)	Mean eff. press. (b)	Cut-off.	Boiler pressure.
1	27	170	160	.73	.86	100.0	53.0	110.0	105.0	121 5	....	....
2.......	42	170	170	.70	.85	106.6	43.0	89.4	98.0	122.0	.71	148
3........	*37	162	160	.70	.73	92.2	44.9	89.8	91.0	119.0	.75	146
4........	*74	173	165	.60	.65	75.6	37.1	74.2	74.9	111.0	....	139
5........	114	170	154	.50	.60	89.4	45.1	91.5	90.5	85.0	....	130
6	*80	162	150	.60	.65	68.3	37.3	74.6	71.4	96.0	.46	146
7........	135	170	138	.50	.60	60.9	35.3	71.7	66.	101.0	.58	140
8........	*128	176	132	.80	.80	65.3	31.7	63.4	64.3	80.0	.46	140
9........	147	170	160	.50	.73	67.5	29.8	62.0	64.7	62.0	.34	130
10	120	162	157	.50	.50	57.2	33.2	83.0	70.0	60.0	.42	130
11	150	168	143	.30	.40	46.7	29.4	59.7	53.2	59.5	.42	135
12........	161	176	162	.30	.40	52.0	26.7	54.5	53.2	53.0	.34	130
13........	161	176	166	.40	.50	62.9	33.6	68 5	65.7	58.5	.33	138
14........	162	177	145	.40	.50	61.4	36.3	73.7	67.5	75.6	.42	128
15........	171	176	152	.20	.30	47.0	23.9	48.8	47.9	53.8	.29	136
16........	180	172	131	.30	.40	48.2	28.9	58.7	53.4	59.0	.38	144
17........	195	170	141	.40	.50	53.9	27.6	56.0	54.9	53.5	.36	136
18........	209	160	147	.50	.73	56.5	26.0	54.1	55.3	52.0	.29	148
19........	230	175	148	.75	....	55.7	24.8	51 6	53.6	49.2	.36	137
20........	272	175	173	.65	....	41.1	19.2	39.9	40.5	43.0	.36	135

* Rev. of simple engine; speed of compound not known.

figures given in the same line necessarily represent the best that either engine can do, and isolated examples form no basis for argument, but a comparison of columns (a) and (b), together with the points of cut-off, clearly shows some of the advantages and disadvantages of the compound as compared with the simple system. It is to be borne in mind that the results given in the table for compound locomotives represent compound working in all cases, and not their capacity when working as simple engines by means of starting or distributing valves. The question of starting power will be discussed subsequently.

The table shows that the maximum average mean pressure of the compound is less than that readily attainable in the simple engine at slow speeds and late cut-offs. For example, in line 2 of the table, the points of cut-off being nearly identical, the compound gave an average mean pressure of 98 pounds, with 170 pounds boiler pressure, as against a mean pressure of 122 pounds for the simple engine, with 148 pounds boiler pressure. The compound is evidently deficient here, and it is necessarily so. The total possible range of pressure in the two cylinders is from 170 pounds to the atmospheric pressure. If there were no losses and the full initial pressure could be maintained throughout the h. p. stroke, the maximum mean pressure, assuming two as the cylinder ratio and an equal division of work, would be two-thirds of 170 in the h. p., and one-third in the l. p. cylinder, or about 113.3 and 56.7 pounds respectively. These are the maximum mean pressures attainable with 170 pounds boiler pressure. It follows directly from this that if it is required that the compound shall have tractive power equal to that of the simple engine in this case, the area of the high-pressure piston would have to be greater than that of the simple engine, in the proportion of about 122 to 100, even though the boiler pressure be increased 20 pounds.

On the other hand, a further inspection of columns (a) and (b) shows that at the earlier points of cut-off and higher speeds, the compound engine is about equal to the simple engine, from which it follows that a high-pressure cylinder of the same size as one cylinder of the simple engine would be sufficient. The correct size for the h. p. cylinder is undoubtedly somewhere between these two limits; for if designed for the power necessary at high speed it will probably be deficient at slow speed and late cut-off; and on the other hand, if made large enough to meet all emergencies, the engine will be over-cylindered for ordinary running. If over-cylindered, the effect will be that, when the only work required of the engine is to maintain the speed on level parts of the road, the necessary mean pressure will be obtained with an earlier cut-off than is advisable with the common forms of valve gear, and as a result of the large ratio of total expansion, the final pressure in the l. p. cylinder will be very low, possibly below the atmospheric pressure. The conclusion which seems evident is, that in designing a compound to take the place of a simple locomotive, the basis for calculation should be, as has been already stated, a detailed record of the work done by the simple engine, taking into account the number and rise of grades, the variation in weights of trains, and the frequency of stops.

The table also shows clearly one of the causes of the economy of the compound locomotive. Comparing lines 11 to 17, inclusive, we find that the average mean effective pressure of the compound engines is 56.5 against 58.7 in the simple engines, the average cut-off being .33 in the h. p. cylinder of the compound and .36 in the simple engine. It appears from this that the compound engine will do the same work with about one-half the *volume* of steam. This is not, of course, a measure of the economy, but it is an indication of one of the causes of the economical performance of compound locomotives.

In the *National Car and Locomotive Builder* of September, 1889, were given a number of indicator cards which serve admirably, in connection with data given in the accompanying description, as an illustration of the application of the principles of design just discussed. Particulars are given of twenty-seven cards, for which the speed ranges from 120 to 312 revolutions, and the mean effective pressures from 71.3 to 41.8 pounds, the average mean effective pressure being 49.8 and the mean cut off about .34. The engine was hauling the New York and Chicago Limited,

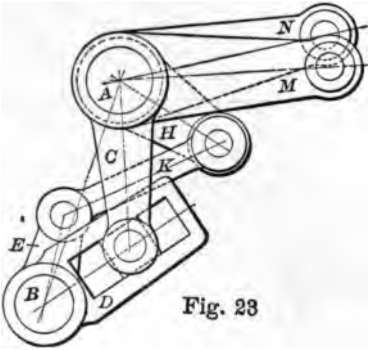

Fig. 23

and only one stop was made between New York and Albany. An examination of the above table indicates that a two-cylinder compound locomotive having a high-pressure cylinder of the same size as a cylinder of the simple engine, and with 170 or 175 pounds boiler pressure, could haul this train under ordinary circumstances without difficulty, the only additional requirement which is apparent being the adoption of some form of starting valve which will insure a starting pressure of about 122 pounds in the h. p. cylinder and its equivalent in the l. p. cylinder. This appears to be a very simple case, the principal remaining question being, how often would the engine be required to develop more power than is shown by these indicator cards? The records of the road should furnish the answer which will indicate whether it is necessary or not to increase the size of the cylinders, and if so, the amount of such increase.

CHAPTER V.

STARTING POWER OF TWO-CYLINDER COMPOUND LOCOMOTIVES.—So much has been written on this subject, and so little has been really logically demonstrated, that it may be well to first briefly investigate the starting power of locomotives of the ordinary form in order to compare the two systems intelligently.

It is the almost universal practice to measure the tractive power of locomotives by applying the formula, $T = \frac{d^2\,p\,s}{D}$ in which d = the diameter of the cylinders in inches, p = the mean effective pressure in pounds per square inch, s = the stroke in inches, D = the diameter of the driving wheels in inches, and T = the tractive power or pull at the rail in pounds. This formula is based upon the fact, that, neglecting friction, the work done in both cylinders during any period, such as one revolution, is equal to that done at the circumference of the driving wheels during the same time. The work done in the cylinders in inch-pounds is 2 × area in square inches × mean effective pressure × twice the stroke in inches $= 2 \times \frac{1}{4} \pi\, d^2 \times p \times 2\, s.$; that at the rim of the driving wheels is the pull in pounds × the circumference of the wheel in inches $= T \times \pi\, D$; therefore,

$$T = \frac{2 \times \frac{1}{4}\, \pi\, d^2 \times p \times 2\, s}{\pi\, D} = \frac{d^2\, p\, s}{D}.$$

It follows from the method of deduction that this formula gives an *average* value for the pulling power, and therefore that, while it furnishes a ready method of comparing the pulling power of locomotives under ordinary conditions, it is of very little use in estimating the starting power, since the *minimum* pull, and not the *average*, is then the measure of the power of the locomotive.

In the ordinary locomotives, assuming that steam can be admitted during the full stroke, and neglecting the effect of angularity of connecting rods, the minimum pull occurs when one crank is on the half center, the other being at a dead point, and the maximum pull is developed when both cranks make an angle of 45 degrees with the center line through the dead points. This can be readily demonstrated by calculation, or by a graphical construction. There are several methods of representing rotative efforts graphically, one of which is shown by Fig. 24, in which

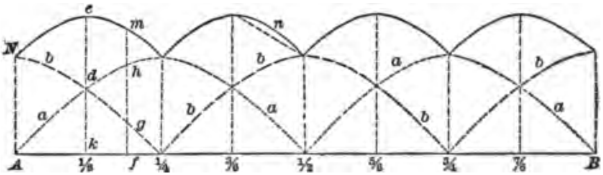

Fig. 24

the dotted line *a* . . *a* represents the rotative effort, or the tangential pull or push, on one crank pin, and *b* . . *b* is that of the other at right angles to it, the steam pressure being assumed as constant throughout the stroke. The method of construction is as follows: Let *A B* be the length of the circumference of a circle, of which *C D*, Fig. 25, is the radius. It can be readily shown that the component *D F*, of the pressure on the piston *D H*, which tends to produce rotation, is proportional to the sine of the angle *a*, through which the crank has turned from a dead point. Divide the line *A B* and the circumference in Fig. 25 into the same number of equal parts. Then through the points of division on *A B* lay off perpendicular distances, such as *k d*, equal to the lines which represent the sines of the angles in Fig. 25, such as *K D*.

The dotted curve *a a* represents the variations in rotative efforts on the crank starting from *C L* during one revolu-

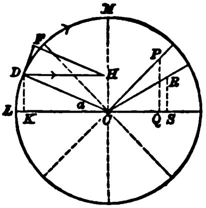

Fig. 25

tion, and the curve *b b*, shown by a broken line, represents the variations in efforts on the crank starting at *C M*, or at right angles with the first. The total rotative effort is shown by the ordinates of the full line curve in Fig. 24, which is obtained by adding the ordinates of the curves for the single crank, for example, $f\,m = f\,g + f\,h$. It is evident that the value of the total effort varies between *A N* and *k e*. In the first case, one crank is on a dead point, and the other is on the half center, or midway between the two dead points. The pull at the rail is then $\frac{1}{4}\,\pi\,d^2 \times p \times s \div D$, which is .7854 of the tractive power as found by the ordinary formula. In the second case the pull is twice that of one crank when making an angle of 45 degrees with the center line, or it is $\frac{1}{4}\,\pi\,d^2 \times p \times 2 \times .707s \div D$, which is 1.11 of the tractive power as usually estimated. It is also clear that there are four maximum and four minimum points during a revolution. These values are determined, as has been said, on the basis that a constant steam pressure can be maintained throughout the stroke, which would be the case in starting if steam could be admitted to the cylinder during the whole stroke. But when the latest cut-off takes place, when the piston is some distance from the end of the stroke, as, for example, at 21 inches with 24 inches stroke, the engine may be in a worse position for starting than that given above as a minimum. When one piston is 21 inches from the beginning of its stroke the other will be about four inches from the beginning of its stroke, and its crank will have turned through about 50 degrees from a dead point. If cut-off takes place at 21 inches, no steam can be admitted to that cylinder during the remainder of the stroke, and the work of starting devolves upon the other cylinder. When the piston has moved four inches from the beginning of the stroke the rotative effort is about three-fourths of the maximum for one cylinder, and is, therefore, about .589 of the tractive power as usually estimated. This corresponds to an ordinate of the curve *a a*, a little to the right of *k d*, and is evidently the most difficult position from which to start the ordinary locomotive. The reduction in the rotative effort on account of the fall in pressure due to the expansion after cut-off and release will be slight. It can be shown on the diagram by laying off radial distances such as *C P* and *C R* on the proper radii to represent the pressures for these crank positions, and using the lines *P Q* and *R S* for ordinates in Fig. 24 instead of those used before. The final effect is shown by the dotted curve at *n*, Fig. 24.

As the locomotive starts the mean effective pressure in the cylinders will be somewhat reduced, but the reduction will not be of large amount within what may be called the starting limits, or until the link would ordinarily be hooked up. As the speed increases the inertia of the reciprocating parts, etc., will be sufficient to modify the form of the diagram of crank efforts, but it is not necessary to consider that in estimating the starting power.

Turning now to the compound locomotive, it is apparent that in the Mallet system the starting conditions are almost identical with those in the simple locomotive. If the h. p. cylinder is of the same size as one cylinder of the simple locomotive, and the cylinder ratio is two, it is only necessary to admit steam of one-half the boiler pressure to the l. p. cylinder in order to have starting power equivalent to that of the simple engine, the same boiler pressure being used. If the l. p. initial pressure is greater than one-half the boiler pressure, the starting power of the compound will be greater than that of the simple engine in all positions in which the l. p. cylinder is available for use in starting, that is, except when the l. p. crank is on a dead point, or when the l. p. valve is in such a position that steam cannot be admitted. If the boiler pressure of the compound is higher than that of the simple engine, and the h. p. cylinder is the same size as one of those of the simple engine, the starting power of the compound engine will be the greater in about the proportion of the two boiler pressures.

In the Worsdell and von Borries type of compound locomotive the conditions in starting are quite different from those just described. When steam is admitted to the receiver by means of the starting valve, the intercepting valve is closed, and the h. p. piston therefore starts against the pressure of the steam or air which filled the receiver just before the starting valve was opened. The amount of this receiver pressure will depend upon the length of time during which the engine has been standing, the condition of valves, etc. If at starting the h. p. crank is at a dead point, the pencil of an indicator, which is applied to the steam end of the h. p. cylinder during the first stroke, will trace a line similar to *a b c*, Fig. 26. The back pressure acting against the other side of the piston during this stroke is shown by a line such as *d e*, the pressure at *e* being somewhat greater than that at *d* on account of the compression in the h. p. cylinder and receiver. The initial back pressure is assumed in the present case as equal to the atmospheric pressure. The diagram, *a b c e d*, thus represents what may be called the effective indicator card for the first stroke of the h. p. piston.

When the h. p. exhaust opens the pressure in that cylinder and the receiver will fall to some point *g*, which can be only approximately determined by calculation. It is located on Fig. 26, by calculation on the basis of no condensation or evaporation during the exhaust. The forward pressure on the h. p. piston during the second stroke will be similar to that during the first stroke, and is shown in Fig. 26 by *h k l*. The back-pressure line during this stroke will consist of, first, a curve *g m*, which represents the compression by the h. p. piston of the steam which fills the space between the h. p. piston and the intercepting valve, until that valve opens; and second, of a line *m n*, of nearly constant pressure, which represents the back pressure during the remainder of the stroke, after the intercepting valve opens and the starting valve is closed. It is generally stated that the pressure of the steam, which is admitted directly to the receiver in starting is reduced by wire drawing to about one-half the boiler pressure. Assuming this to be correct, the h. p. back pressure will become sufficient to open the intercepting valve when about five-eighths of the second stroke has been accomplished, as indicated at *m*, Fig. 26. The net diagram from which the effective pressure on the h. p. piston for the second stroke can be obtained is then *h k l n m g*. A diagram of rotative efforts constructed from these indicator cards is shown in

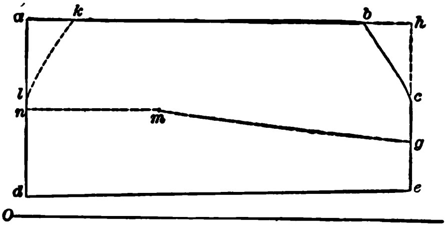

Fig. 26

Fig. 27 by the curve *A E C F B*, from which the reduced effort resulting from the increasing back pressure during the second stroke is apparent.

The distribution of work in the l. p. cylinder in starting does not differ from that in the simple engine. The rotative effort will, therefore, be represented by a curve such as *H K L D M*, Fig. 27, which has the same form as the single crank curves in Fig. 24. The curve in Fig. 27 is constructed on the basis of the initial l. p. pressure, being one-half of the boiler pressure. If the initial pressure is greater than this, the ordinates of the curve between *H* and *K*, *K* and *D*, etc., should be proportionately increased. The combined effort of the two cylinders is shown in Fig. 27 by the full line curve. The intercepting valve opens at about the point *f*, and from that point the engine will work as a compound. It has been already shown that when so working with the customary pressures the power developed at late cut-offs is less than that of the simple engine. The location of the point at which the intercepting valve opens depends upon the pressure in the receiver before starting, the pressure of the steam admitted to the receiver by means of the starting valve, and the size and location of the receiver. For any given combination of conditions it will be found at a definite distance from the point *C*, or from the end of the first stroke of the h. p. piston. In the present case this point was found to be about five-eighths of the stroke from *C*. It is obvious that

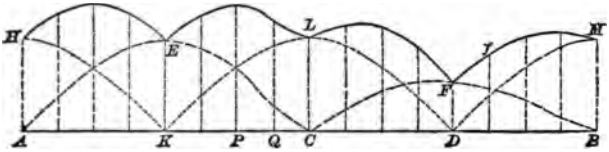

Fig. 27

this action is not at all dependent upon the first stroke of the h. p. piston, but only upon the exhaust from that cylinder. It follows from these considerations that, if the h. p. crank is at a dead point at starting, the engine will move through something over three-fourths of a revolution before compound working begins; but, on the other hand, if the h. p. piston is at the position corresponding to *P*, or near the point at which cut-off takes place, the compound working will begin after about seven-sixteenths of a revolution. If the h. p. crank is in some position such as *Q*, at which the steam valve is closed, the starting must be accomplished by the l. p. cylinder alone; but after a slight movement, sufficient to carry the h. p. crank over the dead point, the cycle will continue as if started at *A*, the effect being to prolong the time of direct working of the l. p. cylinder to about seven-eighths of a revolution.

After compound working commences, and while admitting steam for as much of the stroke as possible, the combined diagram of rotative efforts would be similar to Fig. 24, but with a smaller mean effective pressure, the proportion being with boiler pressures of 170 and 150 pounds in the two types, not greater than 110 to 122, as has been already mentioned. The two diagrams Figs. 24 and 27 are not drawn to the same scale of pressures, but the shape of the full line curves represents with reasonable accuracy the variations in rotative efforts in the simple and compound locomotives. In conclusion, it appears that, with the pressures customary in the two forms, the pulling power of the Worsdell and von Borries' type of compound locomotive in starting may be greater than that of the simple engine having cylinders of the same size as the h. p. cylinder, during the first half revolution approximately, but that after this the power of the compound engine diminishes until it is from eighty to eighty-five per cent. of that of the simple engine.

There is another type of two-cylinder compound locomotive which has not as yet been described in these pages and for which somewhat different conditions exist in starting. In this form the marine practice is followed of placing a valve or cock on the receiver, by which steam from the boiler can be admitted to the receiver and the l. p. steam chest, without the addition of intercepting valves or other complications. This arrangement was applied to a two-cylinder compound locomotive in India by Mr. E. W. McK. Hughes, in 1882, and is in use in Germany in a modified form known as the Lindner starting valve. The latter is illustrated by Fig. 28, which is reproduced from the *Railroad Gazette*. In Fig. 28, *C* is the receiver, *E* is a small pipe connecting the receiver and the main steam pipe, and *J* is the starting valve, which has two ports, *H* and *I*, formed in it at right angles. The lever *K* by which the valve is operated is connected to the reach rod, and the proportions are such that *K* turns through ninety degrees,

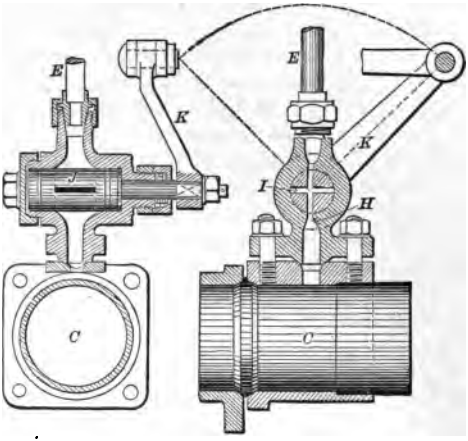

Fig. 28

as indicated in the figure when the reverse lever is moved from one extreme position to the other. The effect is that steam from the boiler is admitted to the receiver when the valve motion is in either the extreme forward gear or the extreme backward gear, and that the cock is closed for intermediate positions. Another feature of the Lindner system is the introduction of two small ports, each having an area of about 0.17 square inch, in the high pressure slide valve, which are so located that when the valve covers the steam port, as after cut-off takes place, that end of the cylinder is connected by means of one of these small ports with the exhaust side of the valve and thus with the receiver. The effect is to admit steam at low pressure to the end of the h. p. cylinder, which is covered by the slide valve, and as the other end is then open to the exhaust and hence to the receiver pressure, the pressure on the two sides of the h. p. piston is partially equalized. In other words, the effective back pressure on the h. p. piston is more or less reduced, so that it offers little resistance in starting. This device is, of course, useful in starting only for the piston positions between full gear cut-off and the end of the stroke.

The possible effect of this arrangement of starting gear will depend upon whether or not a safety valve is provided to limit the maximum pressure in the receiver. If this receiver pressure is equal to one-third of the boiler pressure, with a cylinder ratio of two, the effect of the starting valve is to enable the engine to start with very nearly the same distribution of pressures on the pistons as would be found when it is working as a compound in full gear. The resulting rotative efforts will then be represented by a curve such as the full line curve in Fig. 24, the ordinates or actual pressures, however, being less than those for the simple engine in about the proportion of 113 to 150, with boiler pressures of 170 and 150 pounds.

If the receiver pressure is allowed to become higher than one-third the boiler pressure, the back pressure on the h. p. piston is increased proportionately, and the result is, that the power of the h. p. cylinder is reduced, while that of the l. p. cylinder is increased. The advisability of using the higher pressure depends upon the positions of the cranks at starting. If the l. p. crank is at a dead point, the maximum effort will be obtained by not admitting any steam to the receiver at the instant of starting, but before the engine has made one-eighth of a revolution pressure in the receiver will be necessary to enable the l. p. piston to act. The other extreme is when the h. p. crank is at a dead point in starting. When this is the case, the l. p. crank being then on the half center, full boiler pressure could be advantageously used in the l. p. cylinder, with the result of obtaining a rotative effort about four times as great as in a simple engine starting with the same crank positions. But similarly to the first case, the receiver pressure should be reduced almost as soon as the engine begins to move, or else the h. p. piston will be practically thrown out of action, and the engine might be stalled after making one-fourth of a revolution.

It appears, then, that with this starting valve and a properly loaded safety valve the starting is very simple; but the power is less than that of the simple engine having cylinders of the same size as the h. p. cylinder of the compound, the boiler pressures being 170 and 150 pounds, respectively. With no safety valve, the utility of the device depends upon the position of the crank and the judgment of the engineman, and to this it should be added that the l. p. cylinder, piston, rods, etc., must be made strong enough to bear the full boiler pressure.

It is evident that each of the several types of starting arrangements has decided advantages and disadvantages. With the Mallet type considerable complication seems unavoidable, but the cylinders can be proportioned for the usual work on the road, and the engine will nevertheless be very powerful in starting. With the Worsdell and von Borries type there is also complication, and the engine if proportioned as above will be powerful at the start, but the power rapidly decreases. The Lindner type has the advantage of great simplicity, but the cylinders must be considerably larger than for a Mallet engine of equal starting power.

CONDENSATION IN CYLINDERS.—It is not intended to discuss under this heading the effects of initial condensation and re-evaporation during expansion and during exhaust, but rather to call attention to some of the special requirements of the compound locomotive in this connection. It has been repeatedly shown by writers on compound engines that the loss from condensation should be less in the compound than in the simple engine, under similar conditions, but the gain in this direction may be more than overcome by faulty mechanical arrangement. That is to say, if the cylinders, steam passages and receiver are poorly protected from losses by radiation, the loss by condensation may become sufficiently great to seriously diminish the theoretical thermal advantages of the compound engine.

When steam which is initially dry is expanded in a *non-conducting* cylinder, or without gain or loss of heat, a partial condensation takes place, and the greater the range of pressure is during expansion, the greater will be the condensation. As the transfer of heat between the steam and the cylinder walls is less in the compound than in the simple engine, the conditions in the compound are more nearly those of non-conducting cylinders, while the ratio of expansion is greater, and hence a greater amount of condensation during expansion might be expected. Therefore, if the initial steam is dry in both cases, we would expect to find more moisture in the l. p. cylinder of the compound at the end of the expansion than in the simple engine. This is by no means a complete statement of the case, and is only intended to indicate why the presence of considerable moisture in the exhaust from a compound locomotive is not necessarily evidence of uneconomical working.

In discussing this subject, Mr. von Borries has written substantially as follows: "Since in compound engines re-evaporation is very much lessened—one of the advantages of the system—the steam in the cylinders is always more moist than usual, and in the large cylinder always condenses to some extent. In order that the water may not have an opportunity to re-evaporate, it should be removed. This is most readily accomplished by cutting notches about 2 millimeters (0.08 inch) in width in the cylinder cocks with a sharp-edged file; by this means the cocks are kept sufficiently open to allow the water to escape." "The steam escaping from the stack is always wet; this, however, attends the system, and is no sign of priming, but of the fact that much heat has been withdrawn and converted into work."

In some designs of compound locomotives "safety valves," or, more properly, automatic water valves, are fitted to the l. p. cylinder to prevent damage by possible accumulation of water. This is, however, good practice for all large steam cylinders.

The above remarks simply emphasize the statement already made as to the advisability of carefully lagging the cylinders, and of inclosing the receiver in the smoke-box in order that the steam delivered to the l. p. cylinder may be as dry as possible.

EXHAUST NOZZLES.—There is a very noticeable difference between the exhaust from compound and simple locomotives. As the average expansion in the compound is considerably greater than in the simple engine, the final pressure in the low-pressure cylinder is comparatively low and the intensity of the blast is therefore much reduced. Also, instead of four exhausts in rapid succession during a revolution there are but two, and the total volume of steam exhausted in a revolution is somewhat greater. The result is a more even and less intense urging of the fire, from which more perfect combustion and transfer of heat to the boiler are to be expected, while the quantity of small coal drawn through the tubes will be proportionally less.

The single circular nozzle appears to be the favorite with the designers of compound locomotives abroad, the diameter of the nozzle being in the two-cylinder compound engines concerning which the writer has information, about one-fifth of the diameter of the low-pressure cylinder. The best proportion will naturally have to be determined by experience for different kinds of coal and for the various kinds of service.

SEQUENCE OF CRANKS.—The natural sequence of the cranks in two-cylinder compound locomotives would seem to be to place the low-pressure crank 90 degrees behind the high-pressure crank for forward running. Mr. Urquhart is reported as having tried the effect of placing the low-pressure crank leading in forward motion. The following is quoted from a paper recently read by him in England, as given by the *Master Mechanic*: "From indicator diagrams taken in forward gear with the latter arrangement, it appears that inconvenient and very excessive compression takes place at the first notch in the high-pressure cylinder; so much, indeed, that this notch is not used, but in all other notches a good distribution takes place. In the low-pressure cylinder a much better distribution takes place at all notches compared with engines having the high-pressure crank leading; and this engine with the low-pressure crank leading, which has only recently been put to work, seems to develop more power and to burn less fuel than the others having the high-pressure crank leading. . . . The receiver capacity is equal to that of the high-pressure cylinder, and all the dimensions are the same in both the above cases. The idea of making one engine, by way of trial, with the low-pressure crank leading in forward running originated in its being accidentally noticed that one of the other engines, compounded with the high-pressure crank leading in forward running, seemed to work better and to be more powerful, developing its full tractive force with an earlier cut-off and making steam with greater freedom, when running backward with its train, in which case, of course, the low-pressure crank became the leading one. . . . The only objection the author now sees to the low-pressure crank leading is that it gives too much power in the large cylinder and too little in the small, thus putting the engine out of equilibrium; but this irregularity will in a great measure be obviated by having a receiver about half as large again as those now used on all the present compounds."

The writer has not as yet been able to discover any reason why, with a valve gear which gives an approximately equal distribution for both ends of the cylinders, and with the usual long connecting rods, there should be any practical difference in the steam distribution with the two arrangements of cranks mentioned. Neglecting for the moment the irregularity caused by the connecting rods for any position of the high-pressure crank, the low-pressure is at the same relative point in its revolution whether it is leading or following. For example, if the high-pressure crank is at a dead point, the low-pressure crank will be at either the upper or the lower half center, and the piston and valve will be in the same relative positions, the only difference being that the piston is making the forward stroke in one case and the backward stroke in the other. The same is evidently true for all positions of the high-pressure crank. The relative direction of the stroke will be changed, but the high-pressure piston will be at the same distance from the beginning of a stroke, whether its crank is leading or following. The irregularity in steam distribution caused by the connecting rods must be slight with the long rods which are commonly used in locomotive practice.

It would seem that there must be some other explanation of the difference found by Mr. Urquhart than the sequence of the cranks, and that the real cause has escaped his notice during the short time which this engine, with the low-pressure crank leading, has been in operation.

CHAPTER VI.

ECONOMY OF TWO-CYLINDER COMPOUND LOCOMOTIVES.—The various reasons why compound locomotives should be more economical than those of the common type have been so often stated at length before the railway clubs in this country, before engineering societies abroad, and by the advocates of the different systems of compound locomotive construction, and the whole theory of compound or multiple-cylinder engines has been so thoroughly developed, especially in the numerous books on the steam engine, that an extended discussion of the question in this chapter seems to be uncalled for. Nor does the compound locomotive need the support of arguments to show why it should be more economical than locomotives of the ordinary type, inasmuch as the economy of the compound has been repeatedly demonstrated by that best of all arguments, actual tests in service.

Admitting, then, that a noticeable saving in fuel follows the use of compound locomotives, there remains to be shown the actual amount of this saving, under what circumstances it will pay to build compound locomotives, and the conditions which will make it advisable to alter existing simple locomotives.

The accompanying table contains a list of comparative tests of compound and simple locomotives which would seem to be sufficiently comprehensive in number, duration, boiler pressures, location and kind of service to convince the most skeptical that there is considerable saving in the use of the compound system. The data for this table have been collected from various sources which are presumably reliable. A considerable number of tests are quoted from Mr. Lapage, and for the more recent results in Saxony, I am indebted to Mr. D. L. Barnes. The table shows that the saving in fuel is from 13 to 24 per cent., and it will be noticed that the highest percentage of saving is for a six months' trial (No. 14) of two engines working with the same boiler pressure and of almost identical dimensions. Other cases in which the same boiler pressure was used are Nos. 2, 4, 6, 7, 16 and 19. In No. 7 the boiler pressure was but 120 pounds, which is generally admitted to be too low for the best results ot compound working. The saving in this case was 13.5 per cent., and the next lowest, with equal boiler pressure, is (No, 6) 16 per cent.

It has been claimed by several engineers, whose reputations give weight to their opinions, that a large part of the economy of compound locomotives is due to the higher boiler pressures employed, and that equally good results can be obtained with simple engines. On the other hand, engineers of equal prominence have failed to find any material advantage in the use of what may to-day be called high pressures in ordinary locomotives. There is undoubtedly from theoretical considerations a marked economy in the use of high pressures, provided that full advantage is taken of the greater range of expansion which the higher pressure makes possible. It is not practicable to obtain economical results if a large ratio of expansion is used in one cylinder, on account of the greater condensation, to say nothing of the great variation in the pressure on the crank pin during a stroke. We thus find two conditions which are conflicting as the pressure is increased, and in the locomotive we have the further obstacle that with the ordinary forms of valve gear it is not practicable to cut off steam very early in the stroke. There is no advantage in using higher pressures and correspondingly smaller cylinders, so that the points of cut-off will remain about the same as with the lower pressure, because while the volume used per stroke to obtain the same total pressure on the piston is less, the weight of each cubic foot of steam has increased with the pressure, and the amount of heat drawn from the boiler per stroke will be very nearly the same.

In the compound or multiple-cylinder engine advantage may be taken of the possibilities of high-pressure steam without encountering the losses which follow from extremely early cut-offs in a single cylinder. This is another point of superiority of the compound engine, and while some saving may be accomplished in the simple locomotive with higher pressure by means of somewhat smaller cylinders and a slightly earlier cut-off where practicable, it does not appear that a saving of fuel equal to that of compound engines can be reasonably expected.

The general average of the results in the table, computed by giving each test a value proportionate to the time covered by it, is 18.5 per cent. Making allowance for the probability of special care having been exercised in some of these tests in favor of the compound, it would seem to be safe to estimate upon the basis of 15 per cent. saving. The annual saving per engine on any road on the basis of an assumed percentage, such as 15 per cent., is of course easily calculable from the record of engine mileage and fuel consumption. If the ordinary engine for which it is desired to substitute a compound makes M miles per annum and T miles per ton of coal, which costs D dollars per ton, the annual saving by the compound would be in dollars $\frac{.15 \times M \times D.}{T}$ For example, with coal at \$2 per ton, 30 miles per ton, and 20,000 miles per annum, the annual saving per engine would be $\frac{20.000 \times 2 \times .15}{30}$ $=$ \$200. It is obvious that the advisability of using the compound system varies directly with the *total* cost of fuel per engine per annum, and not necessarily upon the cost of fuel per ton. For instance, with 25 miles per ton, 25,000 miles per annum, and coal at \$1.33 per ton, the annual saving per engine will be the same as that given above.

Mr. von Borries gives the cost of his compound locomotives in one case as from 2 to 3 per cent., and in another case 4 per cent., more than ordinary locomotives of equal weight. He also states that "The power of compound freight locomotives is known by experiment to be 5 to 10 per cent. and of compound passenger and express locomotives 10 to 15 per cent. higher than that of ordinary locomotives of equal weight, so far as not hindered by the adhesion of drivers to rails. For an equal power the compound passenger locomotive is lighter and cheaper than the ordinary locomotive." Accordingly, for equal powers, the first cost is "in freight engines two to five per cent., and in passenger engines eight to twelve per cent. *lower* than ordinary locomotives, similar construction of course being assumed in both cases." Mr. Lapage makes a similar argument in comparing a compound with a simple engine weighing about 36½ tons full. "As there is a saving of from 14½ to 20 per cent. of fuel, the grate area and heating surface may be reduced in proportion, and also the boiler slightly; and the boiler not being so large will bear a somewhat higher pressure, while the weight of water in it will also be less. These items make a considerable difference, for the boiler alone weighs about 8.75 tons, and the water it contains about 2.4 tons, making a total weight of 11.15 tons, of which about 1.5 tons may be saved with the compound. Again, the motion work, springs etc., may be made somewhat lighter, as the strain on the compound is not so great as on the ordinary engine. Without taking this into consideration, however, there is a saving in metal of about .75 ton in the boiler alone." The saving in the weight of the tender, as less fuel and water are required for the same service, is estimated at about two tons, thus making the saving of metal in the engine and tender about three tons. "The cost of the compound for the same power is thus brought to about the same as that of the ordinary engine, or rather less. The engine and tender being lighter and requiring less coal and water, there is a consequent saving in the hauling."

These conclusions are logical, but it does not follow that it will be considered advisable in many cases to reduce the capacity and consequently the weight of the tender, as the increased endurance of the engine with the present tank and coal space may be of more importance. However, the subject of the most economical capacity of tenders for a particular line of service has not in many cases received the attention to which its importance entitles it. On the assumption that the proportions of the boiler of the simple engine which is to be replaced by a compound are satisfactory, including grate and heating surface and steam space, there is a possible reduction in these which is worthy of consideration. To offset this we will have the increased weight of cylinders, pipes, etc., in the smoke-box, and ordinarily some increase in the thickness of boiler plates following the increased pressures. In view of the lack of experimental data of compound locomotives in this country at the present time, it would seem to be advisable to disregard these possible reductions in first cost.

The figures given by Mr. von Borries, from two to four per cent. more than the cost of ordinary locomotives of equal weight, or, on the above basis, for the same work, are presumably based upon the cost of a number of engines, and would not therefore be sufficient for a single engine if the cost of drawings and other expenses incident to getting out a new design are to be charged to it. But even if all expenses which are properly chargeable to the engine are included, it is apparent that the probable saving of 15 per cent. will secure a good return from the investment. It is claimed by the advocates of some of the various systems of compound locomotives that the cost of their maintenance is no greater than that of the ordinary engines. This claim appears to be reasonable, as the slight increase in the cost of repairs which would naturally follow the introduction of additional parts may be balanced by the increased endurance of the boiler, taken all together, on account of the diminished intensity and frequency of the exhaust.

The cost of converting simple locomotives to compound will depend largely upon the type which the working conditions make it advisable to adopt. The simplest case will be that in which one cylinder of the ordinary engine can be retained for the high-pressure cylinder of the compound, and a simple form of starting gear such as Lindner's is adopted. The minimum changes required will then be a new cylinder and piston for the low-pressure, new pipes for the receiver, etc., a starting valve and connections, a safety valve on the receiver, and probably two new slide valves and slight changes in the valve motions. The other extreme is apparently that in which two new cylinders are necessary, and starting and intercepting valves of the Mallet type are adopted. In any case the probable cost can be closely estimated from a detailed cost sheet of ordinary engines of similar dimensions in the shops in which the alteration is to be made. Mr. Hughes gave the cost of converting a 16 × 24 engine in India, which has been already referred to, as about two hundred dollars, which is probably considerably lower than the same work could be done for in this country. Judging from such cost sheets as are at hand, it seems probable that the minimum cost would not be less than three hundred dollars, and may be as high as seven hundred dollars for the more complicated arrangements. But, even with the latter amount as a basis, it is evident that there are many cases in which the expenditure would be advisable.

CHAPTER VII.

THREE-CYLINDER COMPOUND LOCOMOTIVES.

The first question which naturally arises in beginning the consideration of three-cylinder compound locomotives is, what are the reasons why that form should be selected in any case in preference to the two-cylinder type, which is evidently so much more simple? In answer to this, it may be said that the expansion of steam can be carried further in some designs of three-cylinder engines than in the two-cylinder type without using cylinders which are excessively large, and, therefore, that the advantages of high-pressure steam can be more fully realized. The ratio of the volume of the low-pressure cylinders to the high-pressure can be made greater than is practicable with two cylinders, and a more nearly equal distribution of work can, therefore, be obtained. By a proper arrangement of cranks a more uniform rotative power can be secured, and further, a better distribution of weights can be obtained than is possible with the two-cylinder outside connected type.

To avoid possible misunderstanding, attention is called to the distinction between three-cylinder compound engines and triple expansion engines. In the latter the steam performs work in three cylinders in succession, the high-pressure, the intermediate, and the low-pressure. In the former, when released from one high-pressure cylinder the steam is divided between two low-pressure cylinders, or else the exhausts from two high-pressure cylinders unite in the receiver and further expansion takes place in one low-pressure cylinder. In general the name compound is given to those engines in which the steam acts in but *two* cylinders in succession, whether two, three or four cylinders are used, while triple and quadruple expansion mean that the steam is used in three or four cylinders in succession.

The two possible arrangements of three-cylinder compound engines have been applied to locomotives; that with one high-pressure and two low-pressure cylinders by the Northern Railway of France, while that with two high-pressure cylinders and one low-pressure is the arrangement of the well-known Webb compound.

Steam Distribution in Three-Cylinder Compound Locomotives.—The fundamental theory of three-cylinder compound engines does not, of course, differ from that of two-cylinder compound engines. The only differences which exist are the result of the relative angles of the cranks, and are to be found in the variations in the turning moments and in the variations in pressures in the receiver. Each case must be individually analyzed, and the only difference between such analyses and those already given for two-cylinder engines is the greater complication which arises from having three cranks to consider instead of two. As an example of the method to be preferably followed in attempting such an investigation, an arrangement of cranks which has been used for a locomotive is selected. In this form the low-pressure cranks are at right angles and the high-pressure crank makes angles of 135 degrees with them. In the first place we assume the following data: In the high-pressure cylinder, cut-off, .75; release, .90; compression, .90; in the low-pressure cylinders the same distribution.

In Fig. 29 are shown successive positions of the three cranks, h representing the high-pressure crank, L one low-pressure crank, and l the other. Assuming the direction of the revolution to be as indicated by the arrow, an exhaust takes place from the high-pressure cylinder when its crank is at h_1. One low-pressure crank, l_1, is then just commencing a stroke, and the other, L_1, has accomplished about .57 of a stroke, the effect of the angularity of the connecting rods being neglected. From these positions there is free communication between the three cylinders and the receiver until L moves to L_2, where the cut-off takes place in that cylinder, the other low-pressure crank being then at l_2 and the high-pressure crank at h_2. From these positions expansion continues in the cylinder L, while there is still free communication between the other

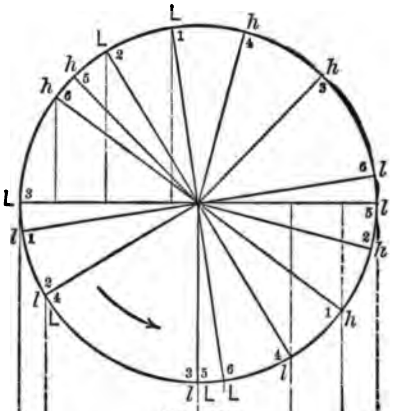

Fig. 29

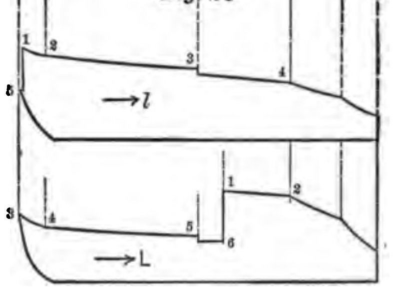

Fig. 30

low-pressure cylinder, the receiver and the high-pressure cylinder until the low-pressure crank *L* arrives at L_3, when steam is again admitted to that cylinder for the re-

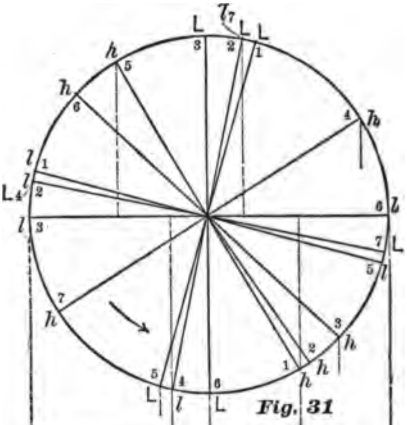

Fig. 31

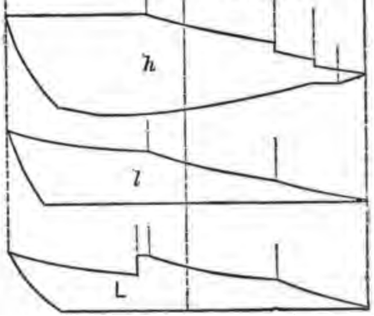

Fig. 32

turn stroke. The other low-pressure crank is then at l_3, and the high-pressure crank is at h_3. All three cylinders are now again in communication, and remain so until the cut-off position l_4 is reached, the other cranks then being at L_4 and h_4. The two cylinders which are represented by h and L remain in communication until the positions numbered $_5$ are reached, when steam is again admitted to the cylinder l. Soon after this the high-pressure exhaust takes place at h_6, and a fresh supply of steam is admitted to the receiver, from which it enters both low-pressure cylinders whose cranks are at L_6 and l_6. These positions correspond to those numbered $_1$, the direction of the piston movement only being changed. It is clear that, when the exhaust takes place from the high-pressure cylinder, the low-pressure piston corresponding to l is always near the beginning of a stroke, while the other is near the middle of its stroke. The effects of this distribution in the low-pressure cylinders are shown in Fig. 30 by indicator cards, which are constructed on the assumption of rapid valve movements and neglecting the irregularities which are caused by the connecting rods. The cards are not drawn to a scale and the variations in pressures are purposely exaggerated. With a relatively large receiver the drop in pressure at l_3 and L_5 will be very small. In practice the readmission at $_1$ would produce a hump in the card L, while the card l would have a form which would apparently indicate that the valve was late in opening.

At earlier points of cut-off somewhat different results will be found. These are illustrated by Figs. 31 and 32, in which it is assumed that cut-off takes place at .4 and release at .75 of the stroke in all three cylinders. Taking the direction of revolution as before, when release occurs in the high-pressure cylinder at h_1, one low-pressure crank is at L_1 and the other is at l_1. A very slight movement brings the crank L to its cut-off position L_2, soon after which steam is admitted to the other low-pressure cylinder at l_3, and that cylinder is in communication with the receiver and the high-pressure cylinder until its cut-off point is reached at l_4. There will then be slight compression in the high-pressure cylinder and the receiver until steam is admitted to the *L* cylinder at the beginning of its next stroke. The remaining events of the revolution are similar to those already noticed and will be made clear by a study of Fig. 31. It will be seen that there is still readmission to the low-pressure cylinder *L*, but that this does not affect the form of the card from the other low-pressure cylinder. With this arrangement of cranks and with the same valve adjustment the indicator cards from the two low-pressure cylinders will be unlike for all points of cut-off. There is in fact but one arrangement of cranks for which the distribution in the low-pressure cylinders will be the same, and that is when the low-pressure cranks are both at right angles with the high-pressure, and therefore either directly opposite each other or parallel. Assuming an equal division of work between the three cylinders, the most uniform turning moment will be obtained by placing the cranks at angles of 120 degrees with each other, but the difference in the distribution in the two low-pressure cylinders will still exist, although it is not probable that this will be of great importance at ordinary running speeds.

An examination of the crank positions for the form of three-cylinder engine having two high-pressure cylinders and one low-pressure cylinder shows similar peculiarities in the distribution. This will be evident from Figs. 33 and 34, which are lettered similarly to Figs. 29 and 31, *H* and *h* representing the two high-pressure cranks, which are at right angles, and *l* the low-pressure crank, which makes angles of 135 degrees with the others. The distribution in Fig. 33 is the same as that in Fig. 29, and that in Fig. 34 is the same as that in Fig. 31. It will be seen that there is readmission to the low-pressure cylinder in both figures; but at the earlier cut-off of four-tenths it is not probable that the effect on a low-pressure indicator card would be noticeable. Placing the cranks at angles of 120 degrees would, as in the first arrangement of cylinders, produce very little change in the indicator cards.

It is evident, from the preceding partial analysis of the steam distribution, that the construction of theoretical indicator cards for three-cylinder compound engines will be considerably more difficult than for the two-cylinder type, but that the same formulas and methods of construction can be used. The remarks which were made in discussing two-cylinder compound locomotives in regard to the effect of varying the capacity of the receiver and the results of changing the points of cut-off are equally applicable to three-cylinder engines. In fact, the only differences are those in the steam distribution, which have been already discussed, and which depend upon the angles made by the three cranks.

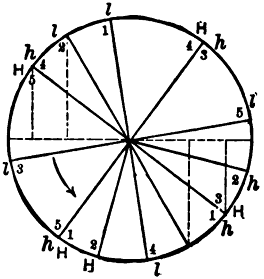

Fig. 33

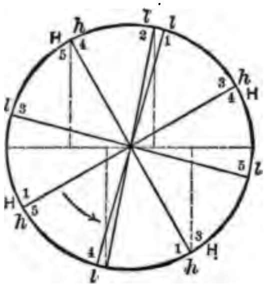

Fig. 34

A mathematical discussion of the three-cylinder type of compound engine, having one high-pressure cylinder and two low-pressure cylinders, and with the cranks placed at angles of 120 degrees with each other, will be found in the appendix to "The Marine Steam Engine," by R. Sennett. The form having two high-pressure cylinders and one low-pressure cylinder does not appear to have been used in marine practice, and its use is not to be expected, inasmuch as one of the chief reasons for using three cylinders instead of two is to avoid excessively large low-pressure cylinders.

In attempting to determine the size of cylinders for three-cylinder compound locomotives, the best guide will undoubtedly be the results obtained with locomotives of that form in practice. When such information is not obtainable, the most satisfactory method will be that advocated under similar circumstances for two-cylinder compound engines, *i. e.*, the construction of, what were called for convenience, theoretical indicator cards, and the alteration of these as experience dictates, to allow for wire-drawing during the opening and closing of valves, drop in pressure, etc. The proportions which appear to have been generally adopted by Mr. Webb are, high-pressure cylinders, 14 inches in diameter; low-pressure cylinder 30 inches in diameter; stroke of all pistons, 24 inches. The ratio of the volume of the low-pressure cylinder to that of both high-pressure cylinders is thus about 2.3. Assuming a mean forward pressure of 175 pounds gauge, in the high-pressure cylinders, and a back pressure in the low-pressure cylinder of 3 pounds above the atmospheric pressure and an equal division of work, we can make an approximate estimate of the maximum power of the engine as follows: The area of the low-pressure piston is 4.6 times that of one high-pressure piston, and, if the work is to be the same in both, the mean pressure in a high-pressure cylinder must be 4.6 times that in the low-pressure cylinder. As the total range of pressure is 172 pounds, and as the mean receiver pressure is approximately the same as the mean high-pressure back pressure and the mean low-pressure forward pressure, we have : 4.6 × l. p. mean effective pressure = 172 — l. p. mean effective, whence l. p. mean effective = 172 ÷ 5.6 = 30.7 pounds. The mean receiver pressure is then 30.7 + 3 = 33.7 by gauge, and the mean effective in the high-pressure cylinders is 175 — 33.7 = 141.3 pounds. A similar calculation can, of course, be made with any assumed mean forward pressure, and this method can also be used for making an approximate comparison of the maximum work done in the cylinders of the three-cylinder compound with that in ordinary locomotives. For example, if the mean forward pressure in the latter is 150 pounds and the back pressure is 3 pounds as before, the total effective pressure during a stroke will be 2 × 147 × area of one piston. To be the equivalent of the compound locomotive this must equal 3 × 141.3 × area of one high-pressure piston. This gives in the present case 221.9 square inches as the piston area of the simple engine, or in other words a simple engine having two cylinders about 16.8 inches in diameter would be equal in power, with the assumed pressures, to the compound engine having cylinders 14, 14 and 30 inches in diameter, the stroke being the same in all cylinders.

The same method can be used to find dimensions for an equivalent three-cylinder engine having one high-pressure and two low-pressure cylinders. If the ratio of the volumes of the two low-pressure cylinders to that of the high-pressure cylinder is 2.3, each low-pressure cylinder will be 1.15 times as large as the high-pressure. Therefore 1.15 × l. p. mean effective pressure = 172 — l. p. mean effective, whence l. p. mean effective = 80 pounds. The mean receiver pressure will be 83 pounds gauge, and the h. p. mean effective pressure will be 175 — 83 = 92 pounds. To find the piston areas we have 92 × area of the high-pressure piston for this engine = 141.3 × area of a 14-inch cylinder, which gives an area of 236.3 square inches, 17.35 diameter, for the high-pressure piston, and 1.15 times this or 271.8 square inches, 18.6 diameter, for each low-pressure piston. An engine having one high-pressure cylinder 17.35 inches in diameter and two low-pressure cylinders 18.6

inches in diameter, is thus equivalent with the assumed pressures to one having two high-pressure cylinders 14 inches in diameter and one low-pressure cylinder 30 inches in diameter. The distribution of work among the three cylinders will be considered in a subsequent chapter.

ILLUSTRATIONS OF THREE-CYLINDER COMPOUND LOCOMOTIVES.—Before discussing other special features of three-cylinder compound locomotives, illustrations of the two forms will be given. The compound locomotive having one high-pressure cylinder and two low-pressure cylinders, which was built by the Northern Railway of France, and to which reference has already been made, was illustrated in *Engineering* of Dec. 6, 1889. The general arrangement of the cylinders and steam connections of this locomotive

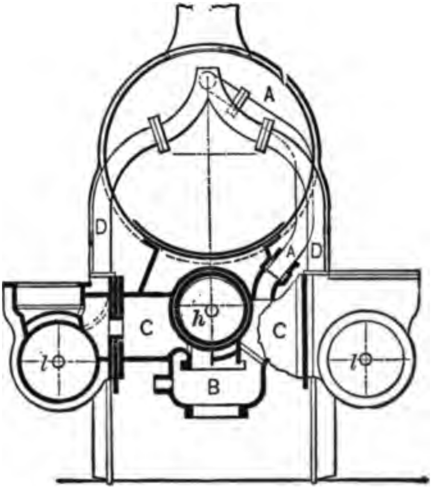

Fig. 35

is shown in Fig. 35. Referring to this figure, *h* is the high-pressure cylinder; *l*, *l* are the low-pressure cylinders; *A* is main steam pipe to the high-pressure cylinder; *C*, *C* is the receiver; and *D*, *D* are the low-pressure exhaust pipes. The low-pressure cylinders are placed as usual and have the valve chests above. The high-pressure cylinder is placed below the smoke-box with its valve chest *B* below it, and is inclined at an angle of one in ten. The locomotive is of the Mogul type, having six coupled driving wheels, the middle axle being the main driving axle for all three cylinders. The low-pressure cranks are at right angles, and the high-pressure crank is midway between them, thus making an angle of 135 degrees with each low-pressure crank. It will be noticed that the receiver is formed in the cylinder castings, and not by pipes, as in the locomotives previously illustrated. The high-pressure valves are a special feature of this engine. These consist of a main valve and a cut-off valve, which slides on the back of or below the main valve, the whole forming a combination which in principle is the same as the Meyer and Ryder cut-offs. The edges of the cut-off valve form an oblique angle with the axis of the cylinder, as in the Ryder valve gear, and the ports in the main valve are correspondingly inclined at the back of that valve, but are twisted so that on the face next to the cylinder they are placed as is customary. The edges of the exhaust port in the cylinder casting are, however, inclined, and the exhaust cavity in the main valve is formed to correspond. The yoke which drives the main valve does not fit it at the sides, and so permits a transverse movement while controlling it longitudinally. A second yoke incloses the valve, and permits a longitudinal movement, but holds it transversely. This yoke is connected to a stem, which passes through a stuffing-box in the side of the valve chest, and is operated from the cab by lever connections. It is clear that the high-pressure cut-off can be adjusted at any time by means of this connection, while the valve is so proportioned that in its extreme position the steam and exhaust ports remain open for all positions of the high-pressure piston, and steam is thus allowed to blow through the high-pressure cylinder without doing work. The engine can, therefore, be started by the low-pressure cylinders with steam from the boiler, the high-pressure piston being then practically inoperative; and as the low-pressure cranks are at right angles, the starting conditions will be the same as for a simple locomotive. The principal dimensions of this locomotive are as follows :

Diameter of high-pressure cylinder (original)......	18.11 inches.
“ “ “ “ “ (present).......	17 “
“ “ low-pressure cylinders (2)........	19.69 “
Stroke of all pistons........................	27.56 “
Diameter of driving wheels.........................	64.9 “
Boiler pressure, gauge...........	199 pounds.
Diameter of boiler, smallest inside.............. ..	52.3 inches.
Tubes, 208, 1.77 inches, length.......................	13 ft. 1.48 inches.
Grate area	22.5 sq. ft.
Heating surface..	1224.9 “
Weight in working order, total............	106,176 pounds.
“ “ “ “ on driving wheels......	90,944 “
Low-pressure valve gear............	Walschaert.
Exhaust nozzle, 4.21 inches in diameter, originally	5.51 inches.

The general arrangement of the cylinders and steam connections of compound locomotives of the Webb system is illustrated by Fig. 36. In this figure *h*, *h* are the high-pressure cylinders, which are placed so that the centers are in a transverse line about four feet back of the front tube sheet, and which are connected to the second pair of driving wheels. The low-pressure cylinder *l* is placed beneath the smoke box and is connected to the forward pair of driving wheels. The course of the steam from the boiler is through the pipes *A*, *A* to *B*, *B*, and thence back to the high-pressure cylinders. The exhaust from these cylinders is led through the pipes *D*, *D*, and thence around the smoke box through two pipes *C* to the low-pressure steam chest. The course of the exhaust from the low-pressure cylinder is clearly indicated in the figure. The disposition of the cylinders and steam pipes is essentially the same in the Webb compounds for passenger and freight service. The most noticeable peculiarity of the system is the absence of driving connection between the high and low pressure axles, there being

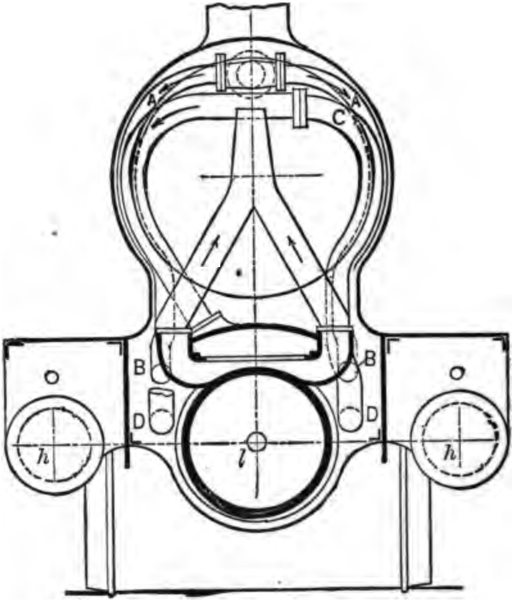

Fig. 36

no coupling rods on engines having two pairs of driving wheels. In one design for freight service there are three driving axles, the first being driven by the low-pressure cylinder, and the second and third, which are coupled, being driven by the high-pressure cylinders. It will be seen that even in this case there is no connection by coupling rods between the high and low pressure cylinders. The principal dimensions of a recent Webb compound locomotive are as follows:

Diameter of high-pressure cylinders (2)	14 inches.
" low-pressure cylinder	30 "
Stroke of all pistons	24 "
Diameter of driving wheels (4)	75 "
Boiler pressure, gauge	175 pounds.
Diameter of boiler, smallest inside	49 inches.
Tubes, 235, 1⅞ in., length	11 ft. 3 in.
Grate area	20.55 sq. ft.
Heating surface	1456.7 "
Weight in working order, total	99,350 pounds.
" " on driving wheels	66,700 "
Valve gear	Joy
Exhaust nozzle	4¾ in. in diam.

CHAPTER VIII.

DISTRIBUTION OF WORK IN THREE-CYLINDER COMPOUND LOCOMOTIVES.—It was shown in the theoretical discussion of the distribution of work between the cylinders of two-cylinder receiver compound locomotives, that with the same points of cut-off in both cylinders and with the ratios of cylinder volumes which are practicable in locomotives, considerably more than one-half of the total work will be done by the low-pressure cylinder. It was also demonstrated that the work can be to a great extent equalized by making the cut-off in the high-pressure earlier than that in the low-pressure cylinder.

The same process of reasoning can be applied to the three-cylinder type of compound engines, inasmuch as we may regard this form as a development of the two-cylinder type, produced by substituting either two smaller high-pressure cylinders for the original high-pressure cylinder, or else two smaller low-pressure cylinders for the original single low-pressure cylinder. It is, therefore, to be expected that, with the same points of cut-off in all three cylinders, considerably more than one-half of the total work will be done in the single low-pressure cylinder of the Webb type of compound locomotive, and in the two low-pressure cylinders of the other form of three-cylinder compound locomotive which, for the sake of brevity, may be called the French type. We may even go a step further and say that, with the ratios of cylinder volumes which are practicable, the total work cannot be so divided that much less than one-half of it will be done in the low-pressure cylinders. This statement is borne out by the published indicator cards of the Webb locomotive and leads to some interesting conclusions.

These indicator cards show that the proportion of the total work which is done in the low-pressure cylinder is from 50 to 65 per cent. at various speeds, with the low-pressure valve in full gear. As making the low-pressure cut-off earlier would increase the proportion of work done in that cylinder, it follows directly that the low-pressure cylinder's share of the total work is *at least* 50 per cent. As the Webb locomotive has no coupling rods between the high and low-pressure axles, and as the weight on each pair of drivers is very nearly the same, it is evident that this division of work is the best under the circumstances. This point and others can be well illustrated by a diagram of crank efforts. Such a diagram is shown by Fig. 37, which was constructed from indicator cards of a Webb locomotive which were published in *Engineering*, May 24, 1889. Steam was cut off at about ten inches in the high-pressure cylinders, and the low-pressure admission was at "full gear." The speed is not recorded, but from the form of the high-pressure admission line is evidently not great. The mean pressure is approximately 81 pounds in the high-pressure cylinders, and about 34 pounds in the low-pressure cylinder, which is equivalent to 34 × 2.3 = 78.2 pounds in the two high-pressure cylinders, the work done in the low-pressure cylinder thus being nearly one-half of the total.

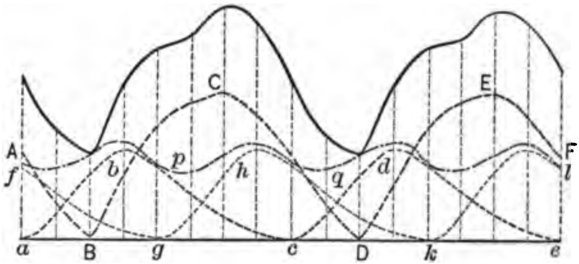

Fig. 37

Referring to Fig 37, *a b c d e* and *f g h k l* show the variations in the turning moments, or the tangential efforts on the cranks, of the two high-pressure pistons, the cranks being at right angles, and the irregularity caused by the connecting rods being neglected. The combined efforts on these two cranks is shown by the curve *f p q l*. The variations in the turning moments on the low-pressure crank are shown by a curve such as *B C D*, and if we assume that the low-pressure crank makes angles of 135 degrees with the high-pressure cranks, this curve and that for the other stroke *D E F A B* will be located as shown in the figure. Combining the high and low pressure diagrams gives the full line curve in the figure which represents the variations in the pulling power of the locomotive during one revolution, as shown by the indicator cards, and therefore without taking the inertia of moving parts into consideration. A comparison of this full line curve with the curve of the combined high-pressure cylinders *f p q l* shows that the angles between the low-pressure and the two high-pressure cranks are not of great importance. If the low-pressure crank were moved back about 25 degrees, so that the maximum moment for the low-pressure crank at *C* would coincide with the minimum for the combined high-pressure cranks at *p*, the combined diagram for all three cranks would be somewhat more uniform, but the difference would not be great. A diagram of crank efforts on the assumption of uniform steam pressures throughout the stroke in each cylinder shows similar peculiarities.

It would seem to be evident from this partial investigation that there would be very little advantage in connecting the high and low-pressure axles of this type of locomotive by coupling rods, excepting possibly for starting, and that, if the suppression of coupling rods is an object of importance, Mr. Webb's design meets the requirement as well as possible with three cylinders. It has been suggested that this type of locomotive might be improved by placing the cranks at angles of 120 degrees and coupling the driving wheels. The effect of this, with the steam distribution and division of work used in the construction of Fig. 37, is shown by Fig. 38, in which the full line curve shows the variations in the combined rotative efforts on the three cranks. It will be seen that the minimum turning moment is greater than that in Fig. 37 and the maximum is less. so that there is a more uniform effort throughout the revolution. The performance of the locomotive at slow speeds would therefore be improved by this arrangement, but as the speed is increased the inertia of the moving parts tends to diminish this apparent advantage, so that

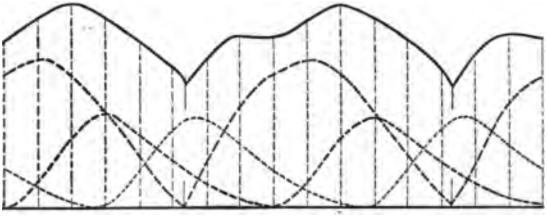

Fig. 38

it is doubtful if there would then be any practical gain by the introduction of coupling-rods.

Turning now to the French type of three-cylinder compound locomotives, it will be found that an application of the same method of reasoning leads to very different results. As has been pointed out, the steam distribution is different in the two low-pressure cylinders, but it is nevertheless to be expected that more than one-half of the total work will be done in the low-pressure cylinders with the same points of cut-off in all three cylinders. Also by adjusting the points of cut-off, the proportion of the total work done in the high-pressure cylinder can be decreased. It is therefore possible with this type of engine to divide the total work equally among the three cylinders.

In this locomotive the two low-pressure cranks are placed at right angles, and the high-pressure crank is placed at 135 degrees with the others. A diagram of crank efforts with this crank arrangement and on the basis of an equal division of work, and steam admission during about seven-eighths of the stroke, is shown by Fig. 39. In this figure, *a b c d* is the high-pressure diagram, and *e f g h k* and *m n o p q* are the low-pressure diagrams. The combined diagram for all three cranks is shown by the full-line curve. If the cranks were placed at angles of 120 degrees, the combined diagram would have the form shown by the full-line curve in Fig. 40, from which it is clear that this disposition of cranks would give a very constant turning moment.

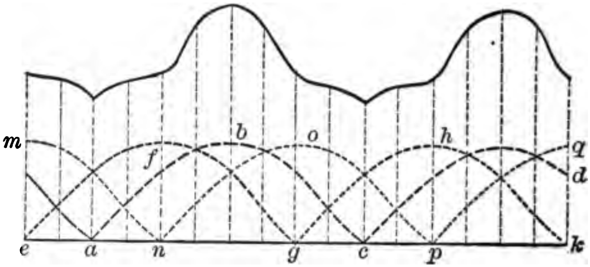

Fig. 39

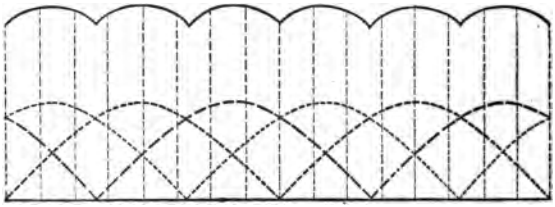

Fig. 40

It follows from these diagrams of crank efforts, in connection with the previous statements regarding the division of the total work, that coupling rods are as much a necessity for the French arrangement of cylinders as they are a useless complication in the Webb system. The work cannot be so divided that as much as one-half of the total will be done in the single high-pressure cylinder of the former, while nearly one-half is necessarily done in the single low-pressure cylinder of the latter.

STARTING POWER OF THREE-CYLINDER COMPOUND LOCOMOTIVES.—It has been already stated that the starting conditions for the French type of three-cylinder compound locomotive are the same as for an ordinary locomotive, inasmuch as the low-pressure cranks are at right angles, and the form of valve gear adopted for the high-pressure cylinder makes it possible to let steam blow through that cylinder without doing work. It is, therefore, possible to admit steam at nearly boiler pressure to the low-pressure cylinders, and the starting power of the locomotive would then be about the same as that of an ordinary locomotive having cylinders of the same size as the low-pressure cylinders, 19.69 × 27.56 inches, and 199 pounds boiler pressure. But as it is not probable that a valve gear of that form would be adopted in American practice, it may not be out of place to consider the effect of some other possible arrangement.

There would not apparently be any practical difficulty in the application of starting gear of the Lindner or the von Borries-Worsdell type to three-cylinder locomotives of this form. With the Lindner starting gear, steam at reduced pressure is admitted to the receiver, and therefore acts as forward pressure on the low-pressure pistons and as back pressure on the high-pressure piston. With an arrangement of cranks such as that of the French locomotive, this amounts to having an ordinary locomotive with 19.69 × 27.56 inch cylinders and with a pressure in the cylinders equal to about one-half the boiler pressure, aided by a single cylinder, 17 inches in diameter, with about the same effective pressure. The position of the cranks, for which the combined turning moment will apparently have its minimum value, is that at which one low-pressure piston has passed just beyond the point of latest cut-off. The high-pressure crank has then moved through but a small angle from a dead point and the other low-pressure crank is not far from the half-stroke position. An approximate calculation, which is sufficient for our present purpose, shows that the combined effort on the crank pins is then equal to that of an ordinary locomotive having 19 × 24 cylinders, at its most disadvantageous position for starting, with 174 pounds effective pressure in the cylinders. As the weights and diameter of driving wheels of the French locomotive do not differ much from those of well-designed six-coupled American locomotives having 19 × 24 cylinders, it is evident that the compound locomotive with the simple Lindner starting gear would be fully equal in starting power to the ordinary locomotive. It would, therefore, seem to be unnecessary to introduce intercepting valves, or other, and more complicated, starting arrangements. That the power of the compound locomotive at other crank positions is superior to that of the other will be made clear by an inspection of diagrams of crank positions, such as Fig. 29.

It has been shown that if the cranks of a French type of compound locomotive were placed at angles of 120 degrees the combined turning moment would be considerably improved, and it appears from an examination of a diagram, such as Fig. 29, that, with a starting gear of the Lindner or von Borries form, the starting power will be increased. As before, the most disadvantageous position for starting will be that at which one of the pistons has passed just beyond the latest cut-off position. The cranks of the other two pistons will then be at approximately 25 degrees and 85 degrees with the center line, and the combined crank effort will be equal to about 1.4 times the maximum of one cylinder as against about 1.13 times that of one cylinder for the most disadvantageous position with the arrangement of cranks actually used on this locomotive. The latter is, of course, the best disposition of cranks with the system of steam distribution which was adopted. It may be useful to note that on the assumption of a uniform steam pressure throughout the stroke on all three pistons inversely proportional to their areas, the combined effort on the crank pins varies from 1.4 to 1.73 times the maximum for a single cylinder for the 90 and 135 degree arrangement, and from 1.73 to 2 times the maximum for one cylinder for the 120 degree arrangement.

In the Webb compound locomotive, as now built, no provision is made for admitting steam from the boiler directly to the low-pressure cylinder. The work of starting therefore devolves at first upon the high-pressure cylinders, which is equivalent to saying that the locomotive is primarily equivalent to a simple locomotive having 14 × 24 inch cylinders and 175 pounds boiler pressure. It has been shown that the turning power on the cranks of the ordinary locomotive in starting may vary from about .75 to 1.4 times the maximum for one cylinder, the lower limit corresponding to the position at which steam cannot be admitted to one cylinder. The tractive power of the Webb locomotive, of which the principal dimensions have been given, would then vary at starting from about .19 to .34 of the weight on the high-pressure driving wheels. It is perfectly possible that the lower value may be encountered in practice, but it is not probable, the case being about the same as that of an ordinary locomotive. The probability is that the high-pressure cranks will be in such positions that the tractive power at starting, with full boiler pressure on the high-pressure pistons, will be from .25 to .34 of the weight on the high-pressure driving wheels. If this is sufficient to slip the driving wheels, steam will then be admitted to the receiver and so to the low-pressure cylinder until the pressure in the receiver rises to such a point that the effective pressure on the high-pressure pistons is not enough to slip the wheels. The locomotive will then be in condition to start as a compound, all three cylinders being available. Another factor has now to be taken into consideration, which is the position of the low-pressure crank. Owing to the absence of coupling rods, this may be anywhere between a dead point, in which position it is of course useless, and a half center where it can exert a pressure equal or superior to the combined high-pressure pistons. It is evident that there is a wide range of possibilities in the starting power of the Webb locomotive, using the term starting power as we have heretofore in discussing two-cylinder locomotives. The exact starting conditions will be partly accidental, and partly dependent upon the skill of the engineer in stopping the locomotive in a good position for starting.

The starting power of locomotives having the Webb arrangement of cylinders, but with the cranks at angles of 120 degrees and coupled driving wheels, will be similar to that of the French type having a like disposition of cranks. Some form of starting valve would then be necessary or else the locomotive would be reduced, for all positions at starting, to a simple locomotive having two small cylinders and cranks 120 degrees apart. Without entering into a detailed investigation, which the practical value of this form of construction does not seem to warrant, it appears that two cylinders would always be available in starting, and that if one of these was the low-pressure cylinder, the locomotive would have ample power, while if the low-pressure was unavailable, there would be an evident deficiency.

THREE-CYLINDER LOCOMOTIVES IN PRACTICE.—The published records of the comparative performance of ordinary and three-cylinder compound locomotives are not as complete as those for locomotives of the two-cylinder type. It is to be remembered, however, that any saving which has been effected by their use is not due to their having three cylinders, but is the result of the application of the compound principle, which necessitates a more economical use of steam in ordinary running. The following is a summarized statement of the work of compound locomotives on the London & Northwestern Railway, as given out by Mr. Webb in June, 1889. Seventy-five compounds had run 11,644,222 miles since the introduction of the system in 1882, with an average fuel consumption of 32.9 pounds per mile. One express locomotive had run 154,342 miles in about 41½ months on an average coal consumption of 36.5 pounds per mile, including that burned while standing in steam or issued for all purposes. The average of forty of the same class was 36.8 pounds per mile. On the Metropolitan District Railway a converted locomotive had run 178,337 miles on an average coal consumption of 23.2 pounds, against 32.4 pounds average for six months for an ordinary locomotive doing the same work.

A report of the results obtained with three Webb compound and three ordinary locomotives in India, doing the same class of work, for the month of Janury, 1886, shows an average coal consumption of 37.57 pounds per mile for the compounds, against 47.14 pounds per mile for the ordinary locomotives. The average mileage for the month was 2,448 and 1,700 miles respectively.

No reports of fuel tests of the French three-cylinder locomotive have as yet been made public. It is reported as having hauled very heavy trains (*Engineering*, Dec. 6, 1889) up grades of 1 in 200 at speeds of from 12½ to 30 miles per hour, but the records do not appear to be valuable for comparison with American practice. It was found that the exhaust nozzle (4.21 inches in diameter) was too small for satisfactory work at the higher speed.

COMPARATIVE SUMMARY.—The reasons which would lead to the selection of the three-cylinder type of compound locomotive in preference to the two-cylinder form have been already mentioned. The better distribution of weights is obvious; it has been demonstrated that the combined turning moment is better, and the greater possibilities in the direction of utilizing high boiler pressures economically are apparent, as we find a cylinder ratio of 2.3 in the Webb locomotive and 2.68 in the French, which would be impracticable ratios for heavy two-cylinder locomotives. To this may be added that with the French arrangement of cylinders it is possible to construct a very powerful and an economical locomotive, for which the low-pressure cylinders will be no larger than those now in common use.

To offset these advantages we will have a cranked axle, a considerable increase in the number of parts and a consequent increase in first cost and expense of maintenance. It is not the intention of the writer to discuss the merits and demerits of cranked axles nor to advocate their use. But in view of the fact that there are hundreds of them in use on locomotives in Great Britain and in Europe, the necessity of using them should not stand in the way of a trial of three cylinder locomotives if other considerations make such a trial advisable.

In this connection it should be noted that the Webb compound locomotive, probably from the fact that it has been the most widely illustrated of any type of compound, has apparently come to be regarded by many as the standard by which all compound locomotives are to be measured. It has been unreasonably attacked and unjustifiably vaunted. The weak features in its design, from an American point of view, have been charged to the compound system in general, and its successes credited to the personal superintendence or "nursing" of the inventor. That the latter is a factor which is at least worth considering in estimating the value of the reported results, should be evident to all who are familiar with the management of steam machinery. It will not probably be questioned that a superintendent of motive power can, by paying special personal attention to some one item of expenditure, secure a notable saving therein, although it is not unlikely that this concentration of supervision may make more important losses possible. And it has been shown that a change in engineers may result in a greater saving of fuel than can ordinarily be expected to follow the introduction of compound locomotives. But there is reason to believe that with the *same handling* and on the *same work* a properly proportioned compound locomotive, regardless of the "system" to which it belongs, will be much more economical in the consumption of fuel than the ordinary locomotive. Whether it has two, three or four cylinders is a secondary matter. Its value as a fuel-saver compared with other means of economizing lies largely in the fact that its success depends less upon the care and attention of the engineer than does that of any device which can be added to the ordinary locomotive. Failures due to errors in design or to unsuitability for the service demanded do not form a sound basis for arguments against the practicability of applying the principles of compound working to locomotives.

CHAPTER IX.

FOUR-CYLINDER COMPOUND LOCOMOTIVES.

There are two general classes of four-cylinder compound locomotives, one of which embraces all designs in which a receiver of large capacity is used, and in which the angles between the cranks are comparatively unimportant, while the other includes only those engines in which the high and low pressure pistons are on one piston rod, or are otherwise rigidly connected, and the " dead space " between the cylinder is reduced to a minimum. The first class includes those locomotives which have two inside connected high-pressure cylinders and two outside connected low-pressure cylinders, and also Mr. Mallet's double bogie compound locomotive. The second class comprises tandem engines and those of the Vauclain type, in which the high and low pressure cylinders are placed side by side and both pistons are connected to one crosshead.

The elementary theory of four-cylinder compound locomotives of the first class is essentially the same as that of two-cylinder receiver engines, and the four-cylinder type may be regarded, as far as the cylinders are concerned, as formed from the two-cylinder type by substituting for each cylinder of the latter two cylinders having a joint volume equal to the corresponding single cylinder. It was shown in discussing two-cylinder receiver engines that, in making approximate calculations to determine proportions, the receiver pressure may be regarded as constant without serious error, assuming that the capacity of the receiver is large compared with that of a high-pressure cylinder. It follows from this that the distribution of work in the cylinders is practically independent of the angle between the high and low pressure cranks when a large receiver is used. If in a four-cylinder engine both high-pressure cylinders exhaust into one receiver, which is the reservoir from which both low-pressure cylinders are supplied with steam, the variations in pressure in this receiver during a revolution will presumably be less than in a two-cylinder engine. We can, therefore, in the design of four-cylinder engines, make use of formulas which are based upon a constant receiver pressure, proceeding at first as if the engine were to have but two cylinders. The formulas are those which are usually given for two-cylinder receiver engines, and are not of special value in the design of two-cylinder compound locomotives on account of the necessity of a very careful analysis of the steam distribution in that type of locomotive if the possible advantages of compound working are to be realized.

In subsequent formulas the letters have the following meaning:

v = volume of high-pressure cylinder.
V = " " low-pressure cylinder.
R = ratio of the cylinders, $V = R\,v$.
r = ratio of expansion in h. p. cylinder.
r' = " " " " l. p. "
p_1 = pressure in h. p. cylinder during admission.
p_2 = pressure in h. p. cylinder when exhaust opens.
p_3 = mean pressure in the receiver.
" = pressure in l. p. cylinder during admission.
p_4 = mean l. p. back pressure.

All pressures are absolute pressures.

Neglecting the effects of clearance, the mean forward pressure in the high-pressure cylinder is

$$p_m = \frac{1 + hyp.\ log.\ r}{r}\,p_1$$

The mean effective pressure is $(p_m - p_3)$ and the work done in the high-pressure cylinder during a stroke is

$v(p_m - p_3)$. Similarly, the mean forward pressure in the low-pressure cylinder is,

$$p'_m = \frac{1 + hyp.\ log.\ r'}{r'} p_3$$

the mean effective pressure is $(p'_m - p_4)$ and the work done in the low-pressure cylinder during a stroke is $V(p'_m - p_4)$.

If the work is to be equally divided between the two cylinders, $v(p_m - p_3) = V(p'_m - p_4)$.

On the basis that volumes vary inversely as the pressures, we have,

$$p_2 = \frac{p_1}{r} = \frac{p_3 V}{r' v} = \frac{p_3 R}{r'}$$

By substituting the value for p_3 obtained from this equation in the preceding one, and reducing, the following is obtained:

$$\frac{p_1}{Rr}\left(hyp.\ log.\ \frac{r}{r'} - \frac{r'}{R}\right) + p_4 = 0. \quad (4)$$

By means of this equation the ratios of expansion in each cylinder (r and r') for which the work done in each will be equal can be determined for any assumed values of p_1 p_4 and R. If it were required that there should be no drop in pressure at the end of the expansion in the high-pressure cylinder, p_2 must equal p_3, from which it follows that r' must equal R. It will be found that equation (4) will give impossible values for r' for many values of r. As r becomes less, or steam is admitted to the high-pressure cylinder during a large part of the stroke, r' will be found to be less than one which is manifestly impossible, and shows that with a late cut-off in the high-pressure cylinder the work cannot be equally divided between the cylinders. On the other hand, as r is made large, r' also increases until it is greater than R, which is an impracticable result, as the receiver pressure would then be higher than the pressure in the high-pressure cylinder at the end of the expansion. For example, if we take $R = 2.3$, $p_1 = 190$ pounds absolute, $p_4 = 20$ pounds absolute, and

$r = 1.33$, or cut-off at 0.75 of the stroke in the high-pressure cylinder, the equation reduces to *hyp. log.* $r' + .4348\ r' = 0.3904$ from which $r' = 0.97$. As steam is admitted during the whole stroke when $r' = 1.0$, it is clear that with the above proportions more than one-half of the total work is necessarily done in the low-pressure cylinder. If r is taken as equal to 4, with the other data the same as before, the value of r' will be found to be 3.75, or the low-pressure cut-off would have to be placed at $1 + 3.75$ or 0.267 of the stroke. But as there will be no drop in pressure between the cylinders when $r' = R$, or when steam is cut off in the low-pressure cylinder at $1 + 2.3 = 0.435$ of the stroke, it follows that to equalize the work in the two cylinders at the earlier cut off the receiver pressure would have to be higher than the pressure at the end of the expansion in the high-pressure cylinder.

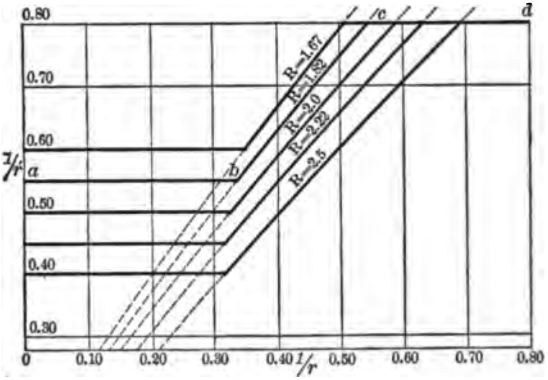

Fig. 41

The engineers of the Paris, Lyons & Mediterranean Railway have applied a formula similar to the above in the determination of the proportions for a class of four-cylinder compound locomotives, and have shown the proper relations existing between the points of cut-off in the high and low pressure cylinders graphically by a diagram similar to Fig. 41. This diagram is reduced from that given in a pamphlet by Mr. C. Baudry, assistant engineer-in-chief of motive power and equipment. Formulas similar to the above will be found discussed at greater length in "Compound Engines," by Mr. Mallet; but in some editions of his book there are many confusing typographical errors. In Fig. 41 the horizontal distances represent the points of cut-off in the high-pressure cylinder, and the vertical distances represent the points of cut-off in the low-pressure cylinder. The inclined lines are curves which represent the solution of equation (4) for different values of R, the pressures used in the construction of the diagram probably being 213 and 21 pounds. For example, if $R = 2.5$ when the high-pressure cut-off is at 0.4, the low-pressure cut-off should be at about 0.5 in order to equalize the work. If the ratio $R \doteq 2$, a cut-off at 0.4 in the high-pressure requires a cut-off at about 0.58 in the low-pressure cylinder. For the cases in which the equation gives values of r' which are too small, the cut-off for the low-pressure cylinder is fixed at 0.8 or the maximum for full gear. For instance, taking $R = 2$, the low-pressure cut-off would remain at 0.8, or full gear for all values of the high-pressure cut-off greater than 0.58, although more than one-half of the work would then be done in the low-pressure cylinder. The other limit to the application of the formula is fixed by making the earliest low-pressure cut-off that at which there will be no drop in pressure between the cylinders. So that finally, the relation between the points of cut-off in the two cylinders is shown by broken lines such as *a b c d*, for which $R = 1.82$. For example, if $R = 2$, the diagram shows that the points of cut-off should vary as follows:

High-pressure...................	.10	.20	.30	.40	.50	.60	.70	.80
Low-pressure.....................	.50	.50	.50	.58	.70	.80	.80	.80

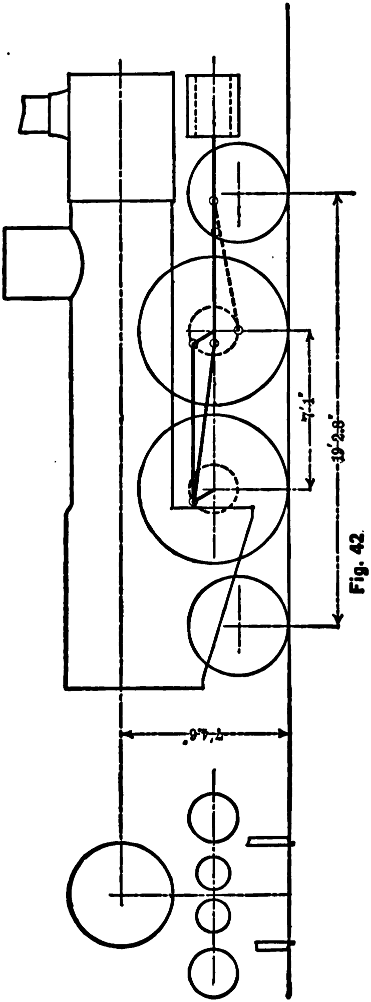

Fig. 42

It does not of course follow that it will be advisable in practice to make use of the early cut-offs given above for the high-pressure cylinder, and it will be remembered that Mr. von Borries has stated, as the result of his experience with two-cylinder compound locomotives, that the most advantageous points of cut-off in the high-pressure cylinder are from 0.3 to 0.4 of the stroke.

Before calling attention to some of the special problems which will present themselves in the design of four-cylinder compound locomotives, illustrations will be given of three experimental types which have recently been constructed by the Paris, Lyons & Mediterranean Railway, and which are described in the pamphlet by Mr. Baudry, to which reference has been made. Fig. 42 shows the general arrangement of the compound locomotives intended for fast passenger service. The principal dimensions of these locomotives, and of the type of simple locomotive which formed the basis for the design, are given in the accompanying table under the heading Fig. 42. The table shows that two compound locomotives of this and of each of the succeeding types have been built, which differ only in the number and diameter of the tubes. It will be seen that in the type of locomotive illustrated by Fig. 42 all four cylinders are placed beneath the smoke-box, with their axes horizontal. The two high-pressure cylinders are between the frames and are connected to the forward driving axle. The low-pressure cylinders are connected to the rear driving axle. The axles are so coupled that the high-pressure crank on each side leads the low-pressure crank on the same side 198°. The object of this arrangement is to obtain as large a value for the minimum starting power as possible. In Fig. 43 is shown the general arrangement of the four-cylinder compound locomotives for freight service. In this locomotive the second driving axle is connected to the low-pressure cylinders, and the third axle to the high-

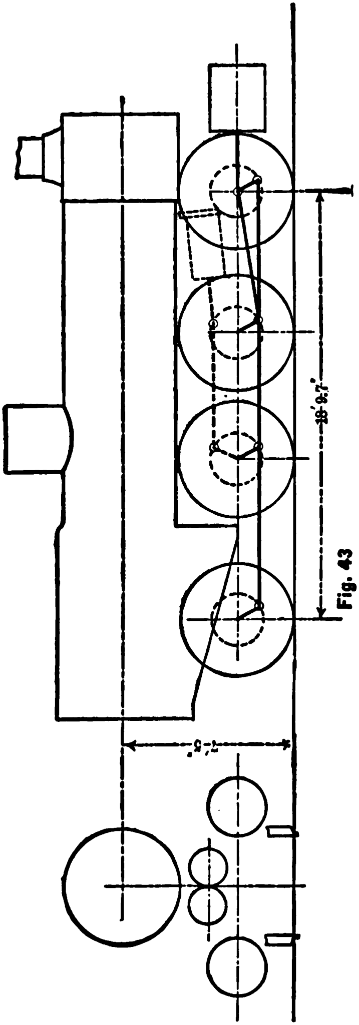

Fig. 43

pressure cylinders. The high-pressure crank on each side leads the low-pressure crank 232° 48′. In the corresponding simple locomotive, of which the dimensions are given in the sixth column of the table, the rear axle is not a driving axle. Fig. 44 illustrates the arrangement adopted for locomotives for steep grades. The high-pressure cylinders are connected to the second axle, and the low-pressure cylinders to the third axle. The high-pressure cranks lead the adjacent low-pressure cranks, as in the other designs, but in this case the angle is 235° 54′.

The Walschaert valve gear is used for all of these locomotives, and the points of cut-off in the high and low-pressure cylinders are adjusted by means of a complicated cam arrangement, designed to fulfill the requirements indicated by Fig. 41. The starting gear adopted for these locomotives consists of simply an auxiliary steam pipe and cock for admitting steam from the boiler to the receiver, which is fitted with a safety valve as usual. In determining the sizes of the cylinders the basis of calculation appears to have been the weight of steam which the boiler may be expected to produce without being forced. This quantity had been found by previous tests to be about 14,500 pounds per hour for the boiler of the express locomotive, and the cylinders should be capable of disposing of this weight of steam. Mr. Baudry states that the steam may, with due regard to the tractive power necessary, be expanded to a pressure of about 44 pounds absolute at a speed of 31 miles per hour, and to about 19 pounds absolute at a speed of 68 miles per hour. The high-pressure cylinders should be capable of utilizing the above weight of steam at slow speeds when the full tractive power of the locomotive is required and also at high speeds. The dimensions selected for the cylinders are thought to meet these requirements, the cut-off in the high-pressure cylinders being of course made earlier as the speed is increased. The method by

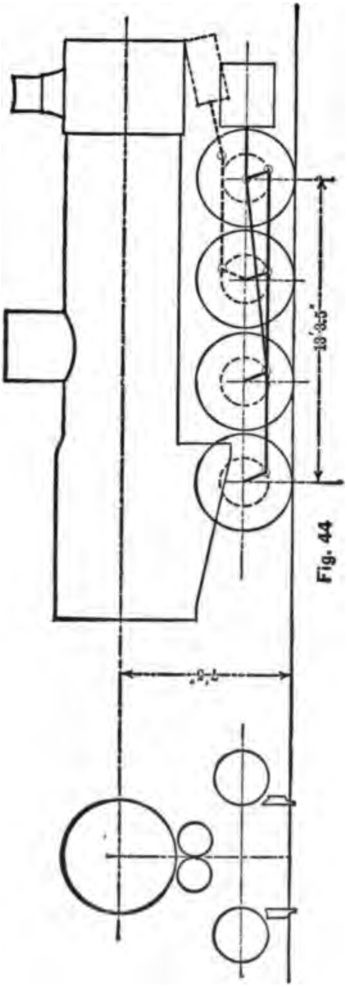

Fig. 44

which the angles between the adjacent high and low-pressure cranks were determined is not explained. Considering the question of balancing the locomotive, the best results would apparently be obtained by placing each high-pressure crank at 180° from the adjacent low-pressure crank. To obtain the greatest value for the minimum turning moment in starting, the angle between the high and low pressure cranks should be 225°, assuming that the pressure of the steam admitted directly to the receiver is such that the total pressure on each piston is the same, and that the cranks on each axle are at right angles. The angles which were adopted appear to be to a certain extent a compromise. For the express locomotive, Fig. 42, in which the question of the balancing of the reciprocating parts at high speeds would be of most importance, the angle selected, 198°, is approximately half way between 180° and 225°. For the freight locomotive the starting power was apparently given greater weight in the problem, the angles in each design being 225° *plus* the angle of inclination of the high-pressure cylinders.

It will be interesting in this connection to note the reasons as given by Mr. Baudry for the adoption of such high steam pressures and the four-cylinder type. "The sole object of the study of new types has been to secure as great economy in fuel as possible. It has therefore seemed necessary to increase the boiler pressure considerably and to adopt the compound system. It is true that some saving in fuel can be realized by the use of the compound system without increase of pressure; but the ultimate advantage seems to be in this case at least questionable. With the pressures of from 128 to 156 pounds per square inch used for locomotives, it is not necessary to resort to this system to obtain a large expansion. Its adoption could then only diminish to some extent the losses resulting from condensation in the cylinders; but as the difference between the extreme temperatures is not very great, and considering the high piston speed, the economy which would result does not seem to be sufficient to compensate for the mechanical complication and the additional expense for repairs which it entails. With high pressures, on the contrary, a little more coal is required to heat the steam, but the possible work of this steam when suitably expanded is increased in a much larger proportion; but the expansion can no longer be accomplished in a single cylinder with slide-valve distribution, the only method which seems to be practicable for locomotives; moreover, the variations in temperature are much greater, and the effect of condensation in the cylinder may become considerable. For these two reasons the compound system becomes a direct consequence of high-pressures in locomotives. It was thus that Mr. Henry was led, without any other reason, to adopt for the new P. L. M. engines the pressure of 213 pounds and the compound system. . . . The employment of four cylinders is not a necessary consequence of the adoption of the compound system. Indeed, the greater part of existing compound locomotives in France and in other countries belong to the Mallet system, and have consequently only two cylinders. But the adoption of this system for locomotives as powerful as those under consideration would have led to very large dimensions, both for the cylinders and for their connections. With four cylinders, acting two by two on different axles, these dimensions are more moderate; there is the further advantage of perfect symmetry, and what is still more important, a notable diminution in the strains on the driving axles and on the rails. The axles are less strained because each transmits but one-half of the work of the locomotive; and the strains on the rails are less since the rapid shocks due to the obliquity of the connecting rods are proportional to the efforts transmitted by the latter, and since these are one-half less when the same work is transmitted by four connecting rods instead of by two."

We are indebted to the engineers of the Paris, Lyons & Mediterranean Railway for their very thorough investigation of four-cylinder receiver compound locomotives, and the report on the performance of these locomotives will be awaited with interest. But the writer is not prepared to admit that all of the points in Mr. Baudry's argument are well taken, or that the advisability of using the four-cylinder receiver type of engine has been demonstrated. A reference to the accompanying table shows that the weights of the compound locomotives are in all cases considerably greater than those of the corresponding simple engines, and they should therefore be more powerful, as they are in fact reported to be. But it does not appear that it was necessary to resort to this type of compound locomotive in order to secure the advantages of compound working. The adhesion weight and the diameter of the driving wheels of the compound express locomotive (Fig. 42) are nearly the same as in the Webb engine and in the Worsdell engine, which have been illustrated in these pages. The total weight of the four-cylinder locomotive is from nine to ten tons greater than that of the two and three cylinder locomotives, and there is, therefore, that much more dead weight. In the designs for the freight locomotives there is more reason for the use of four cylinders, although the three-cylinder compound locomotive of the Northern Railway of France, which we have illustrated, is comparable with the simple locomotive of which the dimensions are given in the sixth column of the table. It certainly cannot be justly claimed that the necessary complication of two-cylinder compound locomotives of the von Borries, Worsdell, or Lindner types, is sufficient to have much weight as an argument against their use. Nor does it follow that pressures as high as 213 pounds per square inch are a necessity in order to secure good results from the use of the compound system. We have no experimental data by which the advisable maximum and minimum steam pressures can be determined, but we have reports of good economical results from two-cylinder compounds working with a steam pressure of but 120 pounds. The economical advantages of compound working are in general more marked as the pressure is increased, but there is obviously a limit beyond which there will be no gain in economy by increasing the pressure. The reasons for this are similar to those which make the adoption of the compound system advisable in any case; briefly, that to secure the advantages of high-pressure steam it must be sufficiently expanded. Therefore, as pressures are increased, a point will be reached beyond which no greater proportion of the energy in the fuel can be utilized with two cylinders of practicable dimensions than can be utilized in a single cylinder with lower pressures. To secure full advantage of the high pressures, triple or quadruple expansion engines must be used as in modern marine practice. It is not, of course, to be inferred that it is advisable to construct triple or quadruple expansion locomotives, although such construction would seem to be more sensible than that of some of the ingenious monstrosities which appear perennially, and which frequently receive the favorable indorsement of men who should know better. We are not prepared to assert that such pressures as 213 pounds per square inch are too high for good work in compound locomotives, but wish simply to call attention to the fact that there are both maximum and minimum limits to the pressures which are properly applicable to compound engines.

The principal advantages of the four cylinder receiver type of compound locomotives appear to be the possibility of obtaining a very uniform turning moment, together with a better running balance than is possible with any other form of locomotive. Whether or not these advantages are sufficient to compensate for the greatly increased number of parts must remain a matter of opinion.

Mr. Mallet's "articulated" four-cylinder compound lo-

[illegible] does not present any special features as far as the steam distribution is concerned. In this form the high-pressure cylinders are fastened to the rear part of the main frames and drive one set of driving wheels, and the low-pressure cylinders are mounted on a front bogie with a second set of driving wheels. There is, therefore, no dead weight and the locomotive is specially adapted for use on very sharp curves. At the same time, as the low-pressure cylinders are on the bogie, the joints in the steam connections are not subjected to high pressure and therefore will not be difficult to keep tight. This system has been used with success on narrow gauge roads, while on the other hand it has been adopted for a locomotive for the St. Gothard Railway, which weighs about 85 tons (2,340 pounds) in working order. There are six axles, coupled in two independent groups, the low-pressure cylinders being placed with three axles on the front bogie, while the high-pressure cylinders are fastened to the main frames and are connected with the other three axles.

DIMENSIONS OF FRENCH FOUR-CYLINDER COMPOUND AND SIMPLE LOCOMOTIVES.

	Fig. 42.			Fig. 43.			Fig. 44.		
	Compound.		Simple	Compound.		Simple	Compound.		Simple
	H. P.	L. P.		H. P.	L. P.		H. P.	L. P.	
Number of cylinders	2	2	2	2	2	2	2	2	2
Diameter of cylinders, inches	12.21	19.68	19.68	13.38	21.26	21.26	14.17	21.26	21.26
Stroke of pistons, inches	24.41	24.41	24.41	25.59	25.59	25.59	25.59	25.59	25.98
Inclination of cylinders	0°	0°	0°	7°48′	0°	0°	10°54′	0°	0°
Maximum travel of valve, inches	4.57	5.59	4.86	5.40	5.63	4.82	5.55	5.63	5.51
Outside lap of valve, inches, front	1.10	1.06	1.22	1 10	1.14	1.10	1.14	1.14	1.14
Outside lap of valve, inches, back	1.02	.95							
Inside lap of valves, inches	0	0	0.04	0.04	0.04	0.04	0.04	0.04	0.02
Maximum admission, per cent. of stroke	71.5	81.5	69.2	76.6	77.9	74.4	77.6	79.3	80.3
Steam ports, area, square inches	10.85	21.7	16.08	13.95	25.11	16.08	14.26	25.11	25.11
Steam ports, per cent. of piston area	9.35	7.13	5.85	10.13	7.09	4.59	9.04	7.09	7.09
	(1)	(2)		(1)	(2)		(1)	(2)	
Number of tubes	185	224	185	247	307	200	247	210	245
Diameter of tubes, outside, inches	1.97	1.77	1.97	1.77	1.58	1.97	1.97	2.17	1.97
Length between tube sheets, feet	13.24	13.24	16.23	14.28	14.28	16.49	13.62	13.62	17.57
Heating surface, fire-box, square feet	125.1	125.1	113.	112.7	112.8	112.3	117.9	117.7	104.5
Heating surface, tubes, square feet	1161.4	1252.6	1422.7	1489.	1643.	1549.	1579.	1491.	2024.3
Heating surface, total, square feet	1286.5	1377.7	1535.7	1601.7	1755.8	1661.3	1696.9	1608.7	2128.8
Diameter of boiler inside largest ring, inches	49.6		49.7	55.1		55.1	59.1		59.1
Area of grate, square feet	25.2		24.1	25.5		23.9	23.5		22.4
Diameter of driving wheels, inches	78.74		78.74	59.06		59.06	49.6		49.6
Weight in working order on driving wheels, pounds	65,260		62,940	125,450		93,120	125,890		120,300
Weight in working order, total, pounds	117,950		111,340	125,450		114,060	125,890		120,300
Ratio of l. p. to h. p. cylinder	2.6		....	2.52		....	2.25		....
Ratio of receiver volume to one h. p. cylinder	5.58		....	4.24		...	3 79		....
Boiler pressure, gauge, pounds per sq. inch	213.3		156.4	213.3		142.2	213.3		156.4

CHAPTER X.

FOUR-CYLINDER COMPOUND LOCOMOTIVES.

Tandem Engines.—The problems to be solved in connection with the steam distribution in compound engines in which the pistons move simultaneously are in some respects quite different from those which are encountered in the design of compound receiver engines. The tandem arrangement is one form of what are frequently called Woolf engines, or continuous expansion engines, the distinctive feature of the class being that the pistons move simultaneously, and that there is no receiver. In the simplest forms of this type, as applicable to locomotives, the high and low-pressure pistons are attached to the same piston rod, and the slide valves of both cylinders are operated by the same link motion. The peculiarities of the steam distribution in this arrangement of cylinders can be best explained by means of theoretical indicator cards, such as Fig. [illegible]. Referring to this figure, *a*, *b*, *d*, *e*, *f*, *g*, *h*, *k*, *a* is the high-pressure card, and *g*, *h*, *l*, *m*, *n*, *q*, *g* is the low-pressure card. In the high-pressure cylinder cut-off takes place at *b*, and there is expansion in that cylinder until the exhaust opens at *d*. There is then a drop in pressure to *e* as the steam in the high-pressure cylinder mingles with that in the passages which connect the cylinders. From *e* to *f* there is further expansion in the high-pressure cylinder and the connecting passages. At *f* the low-pressure steam valve opens and there is another drop in pressure to *g*. From *g* to *h* the cylinders are in communication, and there is expansion until the low-pressure steam valve closes at *h*. From *h* to *k* there is compression in the connecting passages and the high-pressure cylinder,

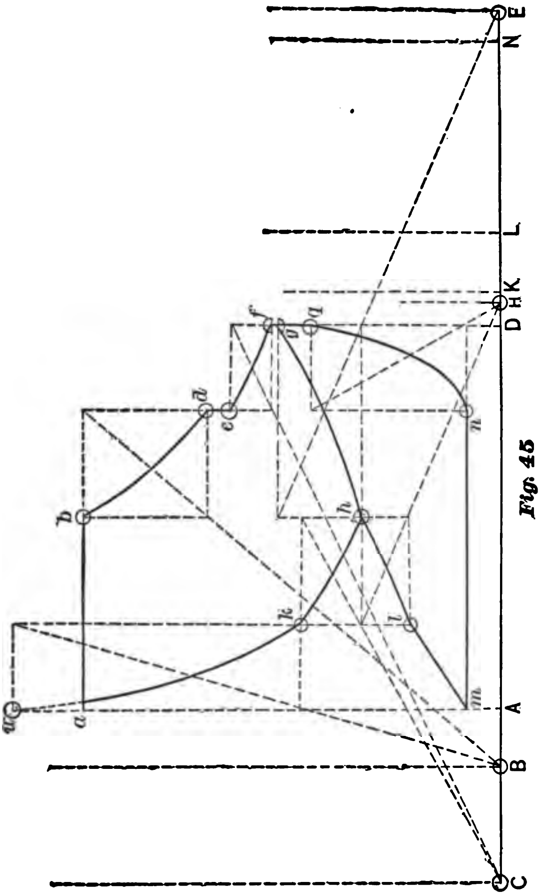

Fig. 45

and when the high-pressure exhaust closes at *k* there is further compression in that cylinder. In the low-pressure cylinder the steam expands from *h* to *l* where release occurs and the pressure drops to the ordinary back pressure line.

The features of this diagram which require special attention are the losses in pressure at *d* and *f* and the compression in the high-pressure cylinder. In order to prevent the drop at *d*, either the pressure in the connecting passages when the high-pressure exhaust opens must be the same as that at *d*, or else the volume of the connecting passages must be practically nothing. The former result can possibly be obtained by adjustments of the low-pressure cut-off, but it is not practicable on account of the unavoidable complications. The only feasible method of reducing this loss to an inappreciable amount appears to be to make the volume of the connecting passages very small compared with that of the high-pressure cylinder. The drop in pressure at *f* can be prevented or reduced by compressing to the pressure *f* in the low-pressure cylinder, or by making the low-pressure clearance very small.

The question of compression in the high-pressure cylinder in this type of engine is even more troublesome than in receiver engines. In order to avoid compressing to a higher pressure than the initial pressure with the usual forms of valve gear, it is necessary that the volume of the high-pressure clearance space should be made large, since the pressure at *k*, where the compression caused by the exhaust closure begins, is unavoidably high. This pressure can of course be somewhat reduced by making the volume of the passages connecting the cylinders large, but, as has been shown, this involves a considerable drop in pressure at *d*. The expedient of giving the high-pressure valve inside clearance may also be employed in connection with a large clearance space to assist in keeping down the compression. In any case in which the shifting link motion is used early cut-offs are to be avoided, both on account of this compression and o avoid the wiredrawing which results from a small port opening.

It is, however, not necessary to resort to very early cut-offs in order to obtain a sufficiently great expansion, as this may be secured by using a comparatively large cylinder ratio. If, for example, we assume a cylinder ratio 3 and a boiler pressure of 160 pounds, a final pressure of about 2.5 pounds above the atmospheric pressure can be secured with a cut-off in the high-pressure cylinder at three-tenths of the stroke at slow speeds, and at high speed a later cut-off would be necessary to maintain the same terminal pressure.

In determining the proportions for the valve gear and the size of the cylinders advisable for a tandem compound which is intended to take the place of an ordinary locomotive, the most satisfactory mode of procedure will be to construct theoretical indicator cards, using the term as we have heretofore, for various points of cut-off, measure the area of these cards or calculate them by the formulas already given in discussing the theory of two-cylinder receiver engines, and finally to adjust or "doctor" the cards for losses, as experience with ordinary locomotives has shown to be necessary. An example of indicator cards constructed in this way is given in Fig. 45, on a much smaller scale, however, than is advisable in practice. The assumed data in this case are as follows: Initial pressure, 175 pounds absolute; cylinder ratio, 3; low-pressure back-pressure, 17 pounds absolute; cut-off in both cylinders, 0.5; release and compression in both cylinders, 0.78; volume of high-pressure clearance, 15 per cent.; volume of low-pressure clearance, 6 per cent.; volume of connecting passages, 0.3 of high-pressure cylinder. The scale of pressures used in the diagram is 80 pounds to the inch. For the benefit of those who may wish to construct such diagrams we will follow through this case in some detail.

The following symbols will be used:

v = volume swept by high-pressure piston.
V = " " " low-pressure "
c = " of high-pressure clearance.
C = " of low-pressure "
i = " of intermediate or connecting passages.

The volumes occupied by the steam at the several lettered points on the diagram are, then,

At b, $= .5\,v + c = .65\,v$.
At d, $= .78\,v + c = .93\,v$.
At e, $= .93\,v + i = 1.23\,v$.
At f, $= v + c + i = 1.45\,v$.
At g, $= 1.45\,v + C = 1.63\,v$.
At h, before cut-off, $= .5\,v + c + i + C + .5\,V = 2.63\,v$.
At h, in l. p. after cut-off, $= .5\,V + C = .56\,V$.
At h, in h. p. and passages after cut-off in l. p., $= .5\,v + c + i = .95\,v$.
At k, before valve closure, $= .22\,v + c + i = .67\,v$.
At k, in h. p. after valve closure, $= .22\,v + c = .37\,v$.
At l, $= .78\,V + C = .84\,V$.
At n, $= .22\,V + C = .28\,V$.

The pressure at d and the curve between b and d may be found by constructing the curve through b with B as the origin, $A\,B$ being .15 of $A\,D$; or by calculation as the pressures may be taken inversely as the volumes, whence pressure at $d = 175 \times .65 \div .93 = 122.3$ pounds. The drop in pressure from d to e depends upon the pressure at k, that in turn depends upon h, and so upon g. The pressure at g depends upon that at q and at f, and so upon e. In any case, there is but one pressure at h which will fulfil the conditions, and that pressure must be determined by calculation. Assuming for the moment that we know the pressure at e to be 112.5 pounds, the pressure at f will be $112.5 \times 1.23 \div 1.45 = 95.4$ pounds. The pressure at g is determined by the mixture of the volume at f at 95.4 pounds with the volume of the low-pressure clearance at pressure q. To find the latter we have pressure at $q = 17 \times .28 \div .06 = 79.3$ pounds. Then pressure at $g =$

$$\frac{79.3 \times .18 + 95.4 \times 1.45}{.18 + 1.45} = 93.7 \text{ pounds.}$$

The pressure at $h = 93.7 \times 1.63 \div 2.63 = 58.1$ pounds. The pressure at $k = 58.1 \times .95 \div .67 = 82.3$ pounds. We can now find the pressure at e which is

$$\frac{122.3 \times .93 + 82.3 \times .3}{.93 + .3} = 112.5.$$

By combining these various expressions for pressures we can readily form a single equation from which the pressure at h can be calculated, which is, in fact, the method by which it was determined in this case.

Having found the pressures at e, g and h by calculation, the various curves of the diagrams can be readily constructed. For the curve between e and f a point C is used for the origin, which is found by laying off $B\,C$ equal to .3 of $A\,D$. The curve $h\,k$ is constructed from the same origin. The compression curve $k\,u$ is laid off from B. To find the origin for the curve $g\,h$, we proceed as follows: At g the steam occupies the volume $v + c + i + C$, and at h the volume occupied is $.5v + c + i + C + (.5V = 1.5v)$. The increase in volume is therefore equal to v, and therefore the scale of this part of the diagram must be such that the horizontal distance from g to h represents v, the volume of the high-pressure cylinder. With this scale of volumes lay off $D\,K = .06V = .18v$, $K\,L = .3v$, $L\,N = v$ and $N\,E = .15v$; then E is the origin from which to construct the curve $g\,h$. For the curves $h\,l$ and $n\,q$ the origin is taken at H, which is found by laying off $D\,H = .06$ of $A\,D$, which for these curves represents the volume of the low-pressure cylinder.

This diagram illustrates the difficulty of keeping the high-pressure compression within reasonable limits. The exhaust closure at 0.78 when cutting-off at .5 is approximately that obtained with the valve gear, of which the distribution was shown in tabular form in the Third Chapter. The valve has 0.25-inch inside clearance, and there is 15 per cent. cylinder clearance. But the pressure at k is 82.3 pounds, and the pressure at the end of the compression u will therefore be $82.3 \times .37 \div .15 = 203$ pounds absolute, or 28 pounds above the initial pressure. To overcome this difficulty it will probably be necessary in most cases to make use of a combination of inside valve clearance, large cylinder clearance and considerable lead.

As a first approximation in determining the size of cylinders the following method may be used. Assume, for example, that a tandem compound is to be substituted for an 18 × 24 simple locomotive, the same boiler pressure of 160 pounds to be retained. With a cut-off at seven-eighths of the stroke, and 160 pounds initial pressure, the mean forward pressure in the high-pressure cylinder will be 173.3 pounds absolute, and the final pressure in that cylinder will be 153.1 pounds absolute. We may assume a drop in pressure of about 5 pounds between the cylinders on account of resistance of ports, etc., so that the initial pressure in the low-pressure cylinder may be taken at 148 pounds. With a cylinder ratio of three, the steam which now fills the high-pressure cylinder and the low-pressure clearance will be expanded to about three times its present volume, and the mean forward pressure in the low-pressure cylinder will be 81.3 pounds absolute. These mean pressures are calculated by the formulas already given in discussing the theoretical distribution in two-cylinder receiver engines. The low-pressure back pressure may be taken at 17 pounds absolute, which gives 64.3 pounds for the mean effective pressure in the low-pressure cylinder. For our present purposes we will take the mean high-pressure back pressure as five pounds above the mean low-pressure forward pressure, which will give a mean effective pressure in the high-pressure cylinder of $173.3 - 86.3 = 87.0$ pounds. If the maximum mean effective pressure which can be developed in the cylinders of the simple locomotive be taken as 155 pounds, we will have, in order that the maximum work of both engines may be the same, assuming that the stroke is the same in both, $87. a + 64.3 \times 3 a = 155 \times 254.5$, in which a is the area of the high-pressure piston and 254.5 is the area of the 18-inch piston. From this equation we find $a = 140.9$ square inches, from which the area of the low-pressure piston is 422.7 square inches. The corresponding diameters are approximately 13.4 and 23.2 inches.

In the above approximate calculation, the effects of early exhaust opening, clearance, compression, and low-pressure cut-off have been neglected, and it will be apparent that a calculation in which these factors were duly considered would be very laborious. But knowing the approximate dimensions of the cylinders the design can be sufficiently completed to determine the volume of ports, passages, etc., and accurate indicator diagrams can then be constructed, by the method already given, by which the final dimensions for cylinders and valves can be determined.

The starting power of a tandem compound locomotive, having cylinders of the dimensions given above, will be about the same as that of the simple 18 × 24 locomotive, when steam at about 82 pounds absolute pressure is admitted directly to the low-pressure cylinders. The steam so admitted acts as forward pressure on the low-pressure pistons, and as back pressure on the high-pressure pistons, so that the locomotive may be said to start as a compound engine. There would, therefore, be no advantage in keeping the starting valve open after the first exhaust from the high-pressure cylinder had taken place. But there is another view of the question of starting which should be noted. If full boiler pressure were admitted to the high-pressure cylinder by the main steam pipe, and the same pressure to the low-pressure cylinder by means of the starting valve, the high-pressure piston would be

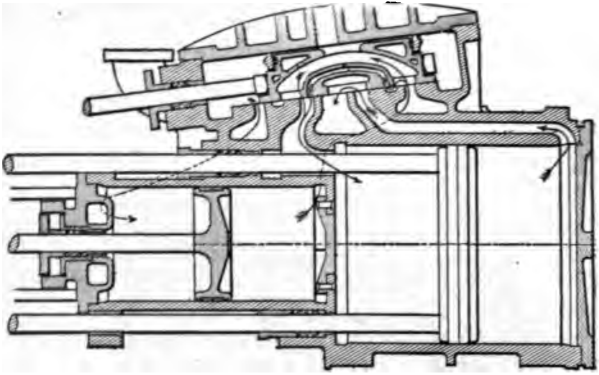

Fig. 46

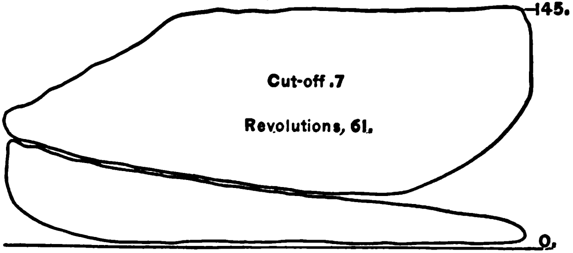

Fig. 47

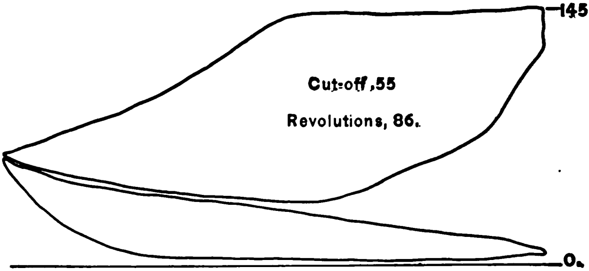

Fig. 48

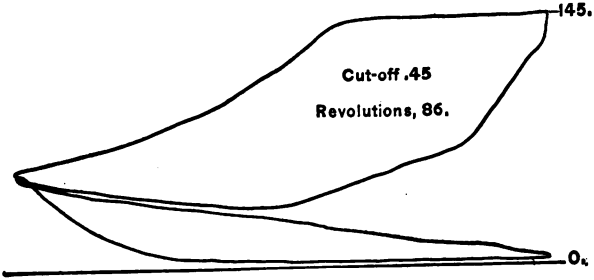

Fig. 49

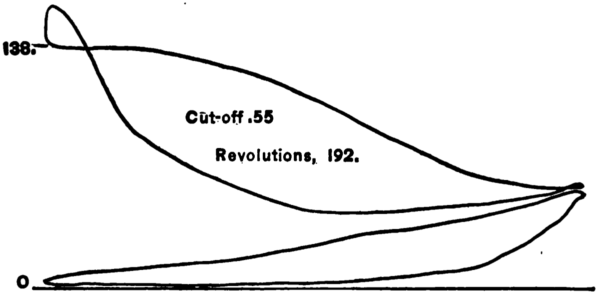

Fig. 50

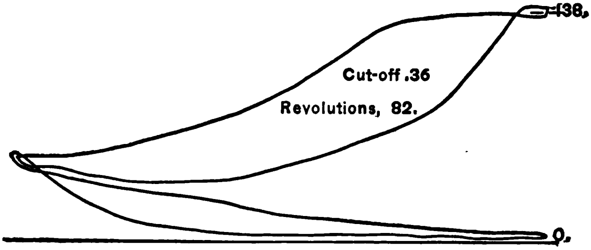

Fig. 51

practically thrown out of action, and the starting power would depend simply upon the area of the low-pressure piston. Therefore, if this piston were but 18 inches in diameter, the possible maximum starting power would be equal to that of the simple engine. Such an engine as this would be deficient in power as a compound, while with an engine having cylinders proportioned for the maximum pressure, as already given, it might be necessary to employ earlier cut-offs for ordinary work on a level than are advisable.

It is probable that the best dimensions for cylinders will be found to be those which will permit the engine to do the greater part of its work at cut-offs of from .4 to .5 in the high-pressure cylinder. It is evident that the only starting gear necessary is an arrangement by which steam can be admitted directly to the low-pressure cylinder for a short time in starting.

A successful application of the tandem form of compound engine to a locomotive has been made by Mr. G. Du Bousquet, of the Northern Railway of France. A detailed description of this locomotive is given in the *Revue Générale des Chemins de Fer* for November, 1888, from which the illustration and indicator cards which follow have been taken. This locomotive is an eight-coupled outside connected engine, all of the weight being on the driving wheels. It was originally a simple locomotive, having cylinders 19.68 inches in diameter by 25.59 inches stroke. The boiler pressure of 142.2 pounds, gauge, is the same as before converting it. The principal dimensions of this locomotive are as follows:

Diameter of high-pressure cylinders.................	15	inches
" " low-pressure "	26	"
Stroke of pistons	25.6	"
Diameter of driving wheels..............................	51.2	"
Total weight, all on driving wheels....................	113,970	pounds
Area of grate...	22.4	sq. ft.
Total heating surface..................................	1,356	"

The changes in the distribution and amount of the weights on the axles on account of converting are given as follows:

	Simple.	Compound.
First axle	26,900	29,670
Second axle	24,470	31,390
Third axle	26,670	30,820
Fourth axle	20,500	22,090
Total	98,540	113,970

To balance the increased weight of the cylinders a foot board weighing 6,600 pounds was put in. Fig. 46 illustrates the arrangement of the cylinders and valve chest, and is worthy of careful examination. It will be seen that the steam distribution for both cylinders is controlled by one valve, the low-pressure valve being, as it were, inside of the high-pressure valve. The arrows clearly indicate the paths of the steam. The principal dimensions relating to this valve gear are as follows:

Travel of valve		6.22 ins.
Steam lap, both cylinders, front		1.34 "
" " " " back		1.22 "
Exhaust lap, high-pressure		0.00 "
" " low-pressure		0.32 "
Ports, high-pressure steam	17.72 ins. ×	1.38 "
" low-pressure "	17.72 " ×	1.97 "
" " " exhaust	17.72 " ×	3.54 "
Angular advance of eccentrics		30 deg.
Clearance, per cent. of cylinder volume h. p.		15.4
" " " " " l. p.		7.0
Volume of connecting passages, per cent. of h. p. volume		16.5

The features of this design which are specially noteworthy are that the dead space between the cylinders is reduced to a minimum, the high-pressure clearance space is large, and that there are no bushings between the cylinders, but instead there are outside stuffing boxes which are easily accessible.

The indicator cards shown by Figs. 47 to 51, inclusive, illustrate the steam distribution in this locomotive. The effect of piston speed upon the distribution is well illustrated by Figs. 48 and 50, which were taken at the same nominal point of cut-off, but as the two pairs of cards are apparently from opposite ends of the cylinders, it is probable that the great increase in compression shown in Fig. 50 is partially due to irregularity in the valve motion. The mean pressures in these diagrams and the percentage of the total work done in the high-pressure cylinder are as follows, the pressures being taken from the tables in the *Revue*, as previously noted:

	Mean pressure.		Per cent. of work done in h. p.
	H. p.	L. p.	
Fig. 47	79.36	30.87	46.2
" 48	63.01	21.76	49.1
" 49	51.20	15.36	52.6
" 50	36.84	15.22	44.7
" 51	31.86	9.53	52.7

This locomotive has been carefully tested in comparison with a simple locomotive belonging to the same original class, and the results are recorded at considerable length in the issue of the *Revue*, to which reference has already been made. The compound hauled trains about 12 per cent. heavier than the simple locomotive, with a noticeable saving in fuel, while with trains of the same weight the saving in fuel, as reported by Mr. Du Bousquet, was from 13.5 to 25.8 per cent. The average of five tests is 21.9 per cent.

CHAPTER XI.

COMPARATIVE SUMMARY.—Having discussed the theoretical principles of the several forms of the compound engine which are applicable to locomotives, and having called attention in detail to the numerous conditions which control the application of these principles, we will now present a summarized statement of the factors which require special care in the design of compound locomotives, and will call attention to some minor points which have not been touched upon in the preceding chapters.

It has been shown that the first requisite in attempting the design of a compound locomotive is detailed information concerning the work which is to be demanded of the engine. This is necessary in the first place, because the chief object in introducing compound locomotives is economy, which can be secured in any engineering work only with full and accurate knowledge of the subject in hand, and in the second place, because, as we have seen, the compound engine is not as flexible as the simple engine, although it is sufficiently elastic to be successfully used for any class of railroad work when originally intended for that service.

In the selection of a type of engine there will generally be considerable opportunity for individual preference. The two-cylinder type is unquestionably the simplest, but the large dimensions of the low-pressure cylinder, for a locomotive, may make the adoption of one of the other forms desirable. If the two-cylinder type is chosen, there is still room for individual opinion in the selection of a form of starting gear and in the determination of the size of cylinders, which will depend to a certain extent upon the nature of the starting mechanism. Assuming that the maximum cylinder power as a compound is to be fully equal to the adhesion under the most advantageous circumstances, the conditions are about as follows : With a starting gear consisting of simply a valve and connections for admitting steam at reduced pressure to the receiver, the locomotive in effect starts as a compound, and the cylinders must be proportioned on that basis. But on the other hand, the cylinders must not be made so large that it will be necessary to cut off very early, or to reduce the initial pressure much by throttling, in those cases in which the total pressure required on the pistons is comparatively small, as, for example, when simply maintaining speed on a level track. The final pressure in the low-pressure cylinder must not be permitted to fall below the atmospheric pressure, and the engine should be proportioned to do the greater part of its work, according to Mr. von Borries, at from 0.3 to 0.4 cut-off in the high-pressure cylinder.

If an automatic intercepting valve is used, the effective pressure per square inch of piston may be increased in starting, but the locomotive will begin to work as a compound after from one-half to one and one-half revolutions of the driving wheels, depending upon the positions of the pistons, and the form and proportions of the intercepting valve, starting valves, and passages. The cylinders must be proportioned to do the maximum work which may be required after compound working begins. If the original Mallet system is adopted, the locomotive can be worked as a simple engine as long as desirable, and the cylinders may be made somewhat smaller than with the other starting devices. This system appears to be specially applicable to converted engines, as one cylinder may be retained for the high-pressure cylinder without necessary increase in boiler pressure.

In two-cylinder-engines the total work can be very nearly equally divided between the cylinders by arranging the valve gear so that the cut-off in the high-pressure cylinder is somewhat earlier than that in the low-pressure. The turning moment on the driving axle will be more nearly constant in a well-proportioned two-cylinder compound than in a simple locomotive, because for the same mean effective pressure the cut-off in each cylinder of the compound will be later than in the simple engine. This possible advantage may be lost by improper valve adjustments which permit excessive compression and a very unequal division of work between the cylinders.

To avoid excessive compression, particularly in the high-pressure cylinder, with the ordinary link motion, inside valve clearance, considerable lead, and large cylinder clearance may be necessary. The question of the proper amount of compression in order to secure the most economical results is one which need not be discussed in this connection, as the problem which confronts the designer of compound locomotives is how to prevent the pressure at the end of the compression from exceeding the initial pressure.

The tandem form of four-cylinder compound locomotive would seem to be next to the two-cylinder type in simplicity. The number of pistons is doubled but the steam distribution on each side of the engine may be controlled by one valve, and there is no increase in the number of connecting-rods, eccentrics, etc. The additional complication is almost entirely in the valves, ports, and cylinders, and not in the driving connections, so that while the first cost will be greater, there is little reason to anticipate much increased cost for repairs. In the design of tandem engines the factors which demand special attention are the probable increase in the weight of cylinders, the compression in the high-pressure cylinder, the drop in pressure between the cylinders, the increased weight of reciprocating parts, and the protection of the cylinders from loss by radiation. The last item is specially important in this class of engines as there is no opportunity for drying out the steam between the cylinders. The division of work between the cylinders is comparatively unimportant. The great advantage of this arrangement of cylinders lies in the opportunity which it affords for the realization of great power together with economy without the employment of exceedingly large cylinders and with but two sets of rods, links, eccentrics, etc. The problem of starting power offers no difficulties in this type of engine.

The three and four cylinder compound locomotives with receivers have many points in common with the two-cylinder form. The sole advantage of the Webb engine appears to be the absence of coupling rods. The three-cylinder type, having one high-pressure cylinder and two low-pressure cylinders, together with the four-cylinder receiver engines, have the advantage of a very uniform turning moment and an excellent balance, with the disadvantage, common to all these forms, of much complication and consequent increased first cost and expense of maintenance.

PISTON SPEED—WEIGHT OF RECIPROCATING PARTS.—It has been urged by opponents of compound locomotives that, while there are admitted advantages in compound working with the "slow piston speeds" found in marine and stationary engines, the high piston speeds found in locomotives make compounding a doubtful experiment. As a matter of fact, there are large triple expansion marine engines running at piston speeds of over 800 feet per minute. The piston speed attained by the quadruple expansion engines of the torpedo boat "Cushing" was 925 feet per minute on her trial, and the speed of pistons of the triple expansion engines of a recent Turkish torpedo boat is given as 936 feet per minute on a trial trip. If these speeds are practicable with triple and quadruple expansion engines, there does not appear to be any good reason for doubting the practicability of speeds of 1,100, or even 1,400 feet, with compound engines. There is undoubtedly a maximum limit to piston speed, and it is lower for compound engines than for simple engines using the same boiler pressure. But the limit is sufficiently high to be comparatively unimportant to the designer of locomotives. The cause which limits the speed is the weight of the reciprocating parts. In an engine working at a speed of 250 revolutions per minute, the reciprocating parts must be started from a state of rest at the beginning of each stroke, and their speed accelerated to about 26 feet per second during approximately a half stroke, which occupies about 0.06 second. The method of calculating the necessary steam pressure per square inch of piston to produce this acceleration need not be explained in these pages, as it does not relate distinctly to compound locomotives. A very full and complete discussion will be found in a paper by Mr. D. S. Jacobus, in Vol. XI. of the Transactions of the American Society of Mechanical Engineers. Taking Mr. Jacobus' figures, the pressure per square inch of piston for a locomotive having a cylinder 18½ inches in diameter and 24 inches stroke, required to overcome the inertia of the reciprocating parts and accelerate them at 250 revolutions per minute, varies from about 55 pounds at 10 degrees from the dead point to 0 at about 80 degrees. The work stored in the reciprocating parts during the first half of the stroke is of course transmitted to the crank-pin during the last half of the stroke, but the effective pressure on the crank-pin during the first half stroke is only that due to the difference between the apparent pressure as shown by the indicator card and that necessary to accelerate the reciprocating parts. It is evident that if the pressure of the steam on the piston is just equal to that required for acceleration at any position of the piston, no pressure will be transmitted to the crank-pin at that point in the stroke, and that if these pressures are equal during the period of acceleration, all pressure which is transmitted to the crank-pin during the stroke will be during the second half stroke. The maximum piston speed of the engine will then be practically not far from that at which this occurs.

The pressure necessary to produce acceleration varies directly as the weight of the reciprocating parts, and as the square of the speed of rotation. The possible means of reducing this pressure are therefore to make the reciprocating parts lighter, or the driving wheels of greater diameter. How much the distribution of pressures on the crank pins will be affected by such changes is a question which must be solved by the designer in each case, and it is a factor which is worth careful consideration, more on account of the crank pin pressures than on account of the limitations of speed. A considerable reduction in weight is effected by the use of steel wherever practicable for the reciprocating parts, and the adoption of the most economical shapes for connecting and coupling rods. In this connection the illustrations of recent two-cylinder compound locomotives in *Engineering* for Feb. 1, 1889, June 21, 1889, and Feb. 7, 1890, are worthy of careful consideration.

ECONOMY.—The principal reasons why a compound locomotive should be more economical than a simple locomotive may be briefly summarized as follows: More economical use of steam by greater expansion; reduced loss from cylinder condensation: better steam distribution on account of later cut-offs, and decreased demand upon the boiler. The commercial economy of the locomotive is the combined result of these causes and also frequently includes a gain by reason of the increased boiler pressure. It is probable that we may add to the above reduced cost of repairs on account of less rapid depreciation of the boiler and more uniform strains on pins and axles. The gain by greater expansion and less loss by cylinder condensation is common to all multiple-cylinder engines, and the reader is referred to works on the theory of the steam engine for detailed discussion of these subjects. Among the many books which are valuable may be mentioned, in addition to those previously referred to in these pages, "The Steam Engine," by Prof. W. D. Marks, and "The Steam Engine," by Prof. J. H. Cotterill.

TESTS IN RUSSIA.—Since the table of reports of comparative tests of two-cylinder compound locomotives given in the Sixth Chapter was prepared, Mr. Thomas Urquhart, Locomotive Superintendent of the Grazi & Tsaritsin Railway, in Southeast Russia, has made public the results of his tests of compound locomotives in a paper read before the Institution of Mechanical Engineers, England, which was published in full in *Engineering* of Feb. 21 and March 7, 1890.

This paper is particularly interesting, as all of the engines tried were converted from simple engines, without increase of boiler pressure. The following is a brief abstract:

An experimental converted engine was put to work in September, 1887, and on trials against a simple locomotive did the same work with a consumption of 22 per cent. less of petroleum refuse. This result was so satisfactory that other locomotives were converted, and in December, 1889, there were in daily use twelve six-coupled freight engines and three four-coupled passenger engines. It is intended to convert the remainder of the locomotives on the line to compounds as fast as the shop facilities will permit. The mean consumption of petroleum refuse per engine mile for freight engines for ten months of 1889 is given in a table as 30.65 pounds for simple locomotives and 25.4 pounds for the compound locomotives. The mean saving in fuel is therefore 17.1 per cent. From April to August, inclusive, the saving by five compound freight engines ranged from 14.75 to 25.57 per cent., the average being 18.96 per cent. A compound passenger engine saved in the same period 18.22 per cent. Mr. Urquhart's conclusion is that 18.5 per cent. may be safely taken as the saving by compound compared with ordinary locomotives. In converting these locomotives one of the original cylinders was retained for the high-pressure cylinder, and the former boiler-pressure of 135 pounds was also retained.

The dimensions of some of these locomotives are as follows: Cylinders, 18½ inches and 25⅝ inches in diameter by 24 inches stroke; six coupled driving wheels, 51 inches in diameter; adhesion weight, 36 tons; tubes, 151, outside diameter, 2⅛ inches; length between tube plates, 13 feet 10⅛ inches; heating surface, total, 1,231 square feet; grate area prior to using petroleum, 17 square feet; Stephenson link motion; maximum travel of valves, 4¾ inches; angular advance of eccentrics, 20 degrees; high-pressure valve, outside lap, 0.81 inch; inside clearance, 0.08 inch; low-pressure valve, outside lap, 0.67 inch; inside clearance, 0.0. The author found that a very small difference in lap, outside or inside, amounted to a great deal in fuel consumption. In more recent locomotives the low-pressure link hanger has been made ⅝ inch longer than the high-pressure hanger.

The starting gear adopted and illustrated is of the Mallet type, being wholly under the control of the engine runner. A cock is provided for admitting steam from the boiler to the receiver, and there is a separate exhaust from the high-pressure cylinder to the atmosphere, the exhaust pipe being placed outside of the smoke box on the high-pressure side. The opening into this pipe is controlled by a plain disk valve, operated from the cab. A flap valve prevents the steam in the receiver from escaping back through this pipe or to the high-pressure cylinder, and is connected so as to be worked by the same lever as the above-mentioned exhaust valve.

It was found by trials that these converted locomotives were capable of hauling as heavy trains as the simple locomotives. With the reverse lever in the same notch in both classes of engines, the power of the compound was found to be less than that of the simple engine, the difference increasing as the cut-off was made earlier, from 5 per cent. in full gear to 28 per cent. in the "first notch" in the freight engines. The volume of the receiver was made equal to that of the high-pressure cylinder in the earlier engines and in the later designs was increased to 1.3 and then to 1.8 times the volume of the high-pressure cylinder, with good results.

It may be well to note that there is nothing contradictory in these results of Mr. Urquhart's experiments to what has already been said concerning the power of two-cylinder locomotives. With the type of starting gear adopted, the engine cannot be deficient in starting power if originally well proportioned, and after starting there is no difficulty. The mean pressure with any given cut-off in the high-pressure cylinder is necessarily less than in the simple engine with the same cut-off and the same boiler pressure, since the expansion is greater in the compound.

Mention was made in a previous chapter of the reported advantage found by Mr. Urquhart in arranging the cranks so that the low-pressure led in forward motion, and the writer stated that the reason for the improvement in the steam distribution which was claimed was not apparent. The tables in Mr. Urquhart's paper show that in the two engines which were compared—one with the low-pressure crank leading and the other with the high-pressure leading—one high-pressure cylinder was 18.5 inches in diameter while the other high-pressure cylinder was 18.11 inches in diameter, for the same notch in the quadrant the points of cut-off in the high-pressure cylinders were different, and in one engine the low-pressure cut-off was considerably later than the high-pressure, while in the other the distribution was very nearly the same in both cylinders. These points of difference are sufficient to vitiate the conclusions in regard to sequence of cranks. The only conclusion which is apparent after a study of crank diagrams is that the sequence of cranks is of very little consequence.

The following, from *Engineering*, referring to Mr. Urquhart's tests, is worth noting : "His experience is especially valuable as the steam pressure has not been increased in the converted engines, and thus there is no fear that it may be argued the economy claimed for the compounding has been partially, if not wholly, due to another cause, *i. e.*, the higher pressure. This is a line of argument with which marine engineers were very familiar at one time. Mr. Urquhart's experience is also valuable for another, and perhaps more, important reason. It is often claimed by those who do not believe in compounding locomotives that the economy in fuel stated to be made is due more to the firing than to the system. . . . But Mr. Urquhart's locomotives are mechanically stoked—that is, they are worked with liquid fuel—so that the ordinary conditions of good or bad firing do not apply. Whether the 'human factor' crops up in other directions we do not know. . . . The evidence in favor of the compound locomotive, as being economical in fuel, is now of a very complete nature, and the 'human factor' argument is the chief one left to those who oppose the system."

CHAPTER XII.

AMERICAN PATENTS.—There are apparently but few United States patents which bear on the subject of compound locomotives, and most of these are of very recent date. The following list includes all which the writer has been able to find which appear to be of interest to designers of compound locomotives, although it is not improbable that others exist, carefully hidden by the erratic style of indexing adopted by the Patent Office. No attempt has been made to determine the breadth of the claims in any case.

H. D. Dunbar, No. 264,937, Sept. 26, 1882; 276,368, April 24, 1883. Slide valves for tandem compound engines.
E. G. Davis, No. 274,571, March 27, 1883; 279,544, June 19, 1883. Slide valve for tandem compound engines.
W. B & J. A. Johnson, No. 351,921, Nov. 2, 1886. Four-cylinder compound locomotive, with one receiver connected to all four-cylinders.
T. W. Worsdell, No. 360,834, April 5, 1887. Compound engine with intercepting valve, substantially as previously described.
A. von Borries, No 361,471, April 19, 1887. Compound locomotive with intercepting valve, substantially as previously described.
R. Lindner, No. 404,295, May 28, 1889. Starting gear for compound locomotives, substantially as previously described.
R. H. Lapage, No. 405,569, June 18, 1889. Compound engine with intercepting and starting valve.
R. H. Lapage, No. 405,570, June 18, 1889. Bogie compound locomotive with high pressure cylinders on main frames and low-pressure cylinders on a bogie truck,
S. M. Vauclain, No. 406,011, June 25, 1889; 406,012, June 25, 1889. Four-cylinder compound locomotive with one or two piston steam-distributing valves.
A. J. Pitkin, No. 417,083, Dec. 10, 1889. Two-cylinder compound locomotive with piston intercepting valve.
F. W. Dean, No. 433,164, July 29, 1890. Compound engine.

THE DUNBAR SYSTEM.—A four-cylinder compound locomotive was built by the Boston & Albany Railroad Company in 1883, under the Dunbar patents. The cylinders were 12 inches and 20 inches in diameter, by 26 inches stroke, and were arranged tandem with the high-pressure and low-pressure pistons on the same piston rod. The engine could be worked compound or non-compound at will. After working about seven months the locomotive was changed to a simple engine as it was apparently no more economical than the simple locomotives. It is stated that the ports were too small and that the inventor was absent during the trial. As the locomotive was an experiment it is not surprising under the circumstances that the results were unsatisfactory.

THE PITKIN TYPE —A two-cylinder compound locomotive designed by Mr. A. J. Pitkin and built at the Schenectady Locomotive Works was put to work on the Michigan Central Railroad in September, 1889. The general arrangement of the cylinders and steam connections of this locomotive is shown by Fig. 52. The distinctive feature of the engine is the intercepting valve, which is shown by Fig. 53 which is a plan of the bushing which incloses the valve and by Fig. 54 which is a vertical section through the valve, bushing and saddle.

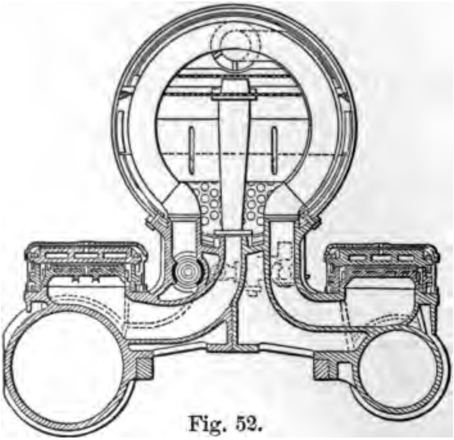

Fig. 52.

The valve is shown in the position which it occupies in

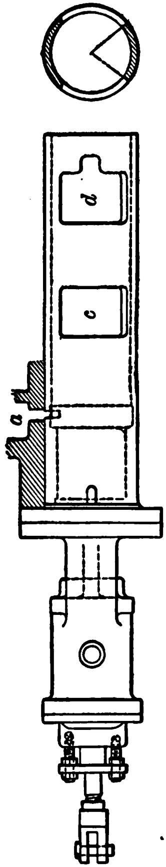

Fig. 53.

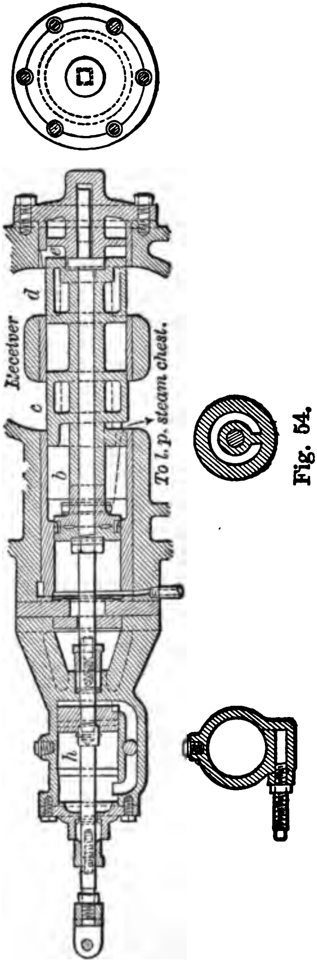

Fig. 54.

starting; that is, before compound working begins. In this position the ports *c* and *d* are closed by the intercepting valve and the connection between the low-pressure steam chest and the receiver is thus cut off. The small port *a* (Fig. 53) is connected by a pipe and a pressure-reducing valve to the high-pressure steam pipe. By this means steam at reduced pressure is admitted to the space *b* and thence, as indicated by the arrow, to the low-pressure steam chest. As the parts of the valve on either side of *b* are of different diameters, the pressure in this space tends to hold the valve in the position shown in Fig. 54. When the locomotive starts, the high-pressure cylinder exhausts into the closed receiver, and the back pressure thus created acts upon the forward end of the intercepting valve by means of the passage shown at *e*. The pressure in the receiver rapidly increases until the total pressure on the forward end of the valve is sufficient to overcome the total effective pressure at *b*, when the valve is forced to the back end of its stroke, the direct steam supply to the low-pressure cylinder is cut off and compound working begins. To prevent the valve moving too rapidly a dash-pot in the form of an oil cylinder, *h*, is added. The valve stem is continued through this oil cylinder and is connected by levers to an index in the cab which indicates the position of the valve.

The locomotive fitted with this intercepting valve has now been at work for several months with apparently satisfactory results. While its fuel saving capabilities have been demonstrated, it has not been thoroughly tested against a simple locomotive of proper dimensions to make a satisfactory comparison possible. The dimensions of this locomotive are as follows:

Diameter of h. p. cylinder	20 in.
" l. p. "	29 in.
Stroke of pistons	24 in.
Ratio of cylinder volumes	2.1
Size of steam ports, h. p	18 in. × 1⅛ in.
" " " l. p.	20 in. × 2⅛ in.

Size of exhaust ports, h. p.	18 in. × 3 in.
“ “ “ l. p	20 in. × 3 in.
Maximum travel of valves	6⅛ in.
Lap of valve, h. p., outside	1 ⅝ in.
“ “ “ inside	— ⅜ in.
Lap of valve, l. p., outside	1⅛ in.
“ “ “ inside	— 1-16 in.
Lead, full gear	5-32 in.
Number of driving wheels	6
Diameter of driving wheels	68 in.
Style of boiler	Wagon top.
Diameter of shell outside first ring	58 in.
Thickness of plates, steel	9-16 and ½ in.
Size of firebox, inside	length 96 3-16 in.; width 42⅞ in.
Tubes, 247, 2 in.; length over tube sheets	12 ft.
Heating surface, tubes	1,540.3 sq. ft.
“ “ firebox	137.1 sq. ft.
“ “ total	1,677.4 sq. ft.
Grate surface	28.5 sq. ft.
Total weight in working order	126,800 lbs.
“ “ on driving wheels	97,000 lbs.
Total wheel base, engine	22 ft. 6 in.
Driving wheel base	12 ft. 2 in.
Rigid wheel base	6 ft. 3 in.
Total wheel base, engine and tender	48 ft.

As built, the high-pressure valve had neither inside lap nor clearance; the low-pressure valve had ⅛ inch inside lap, and the lead of both valves was $\frac{3}{32}$ inch. These proportions have been changed to those given in the table of dimensions. In addition to this, the volume of the receiver has been increased from that of the high-pressure cylinder to 1½ that volume, and the high-pressure clearance has been increased from 8½ to 10 per cent. of the piston displacement. These changes were made after the indicator had demonstrated their necessity, and it will be seen they are entirely in accord with the proportions already recommended in these pages. Other locomotives of this type are now being constructed.

THE VAUCLAIN TYPE.—A locomotive of this type was built by the Baldwin Locomotive Works in the fall of 1889, and was put to work on the Philadelphia Division of the Baltimore & Ohio Railroad. The general arrangement of the cylinders and valve is shown by Figs. 55 and 56. Referring to Fig. 55, *h* is the high-pressure cylinder, *l* is the low-pressure cylinder, and *v* is the valve bushing. The method by which the power from both cylinders is transmitted through one cross-head is shown in Fig. 56, which

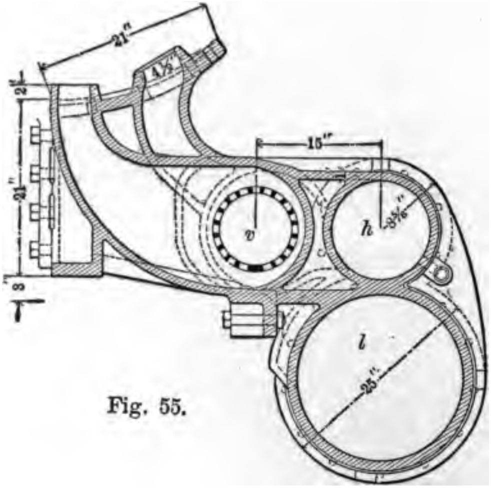

Fig. 55.

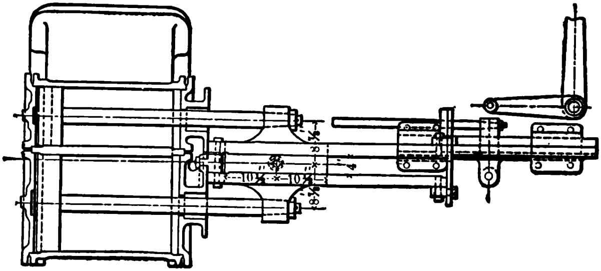

Fig. 56.

also shows the direct connections of the valve. The steam distributing valve is a hollow piston valve, the action of which is illustrated by Fig.

57, which represents a longitudinal section through the valve and the high-pressure cylinder. Steam from the boiler enters the steam chest through the ports *A*, *A*, and passes thence by the port *B*, as indicated by arrows, to one end of the high-pressure cylinder. The exhaust from this cylinder passes through the valve to the port *C*, and thus to the low-pressure cylinder. The exhaust from the low-pressure cylinder passes around the outside of the central portion of the valve, as indicated, to the exhaust pipe.

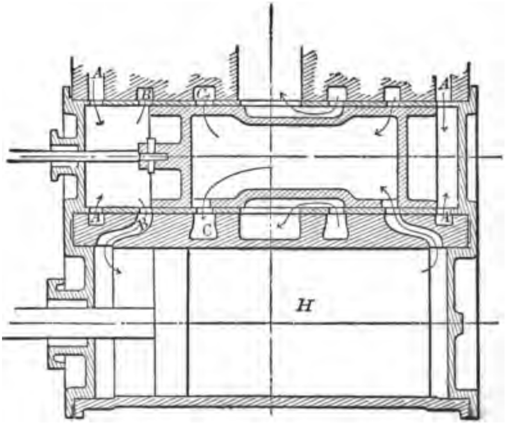

Fig. 57.

The method of distribution thus resembles that with the Dudgeon slide valve, which was at one time used for marine compound engines.

The feature of this design which at first glance would seem to be most open to criticism is the connection to one cross-head of two pistons of which the centers are about 18 inches apart and on which the total pressure must vary considerably. To determine the amount and variation of this difference of pressure with reasonable exactness an examination of a large number of indicator cards taken simultaneously from both high and low pressure cylinders would be necessary. Some knowledge of the subject can, however, be gained from the examination of a few cards, and for this purpose the indicator diagrams shown in Fig. 58 have been selected from a blue print sheet furnished by the builders of this locomotive. The data for these diagrams are given as follows:

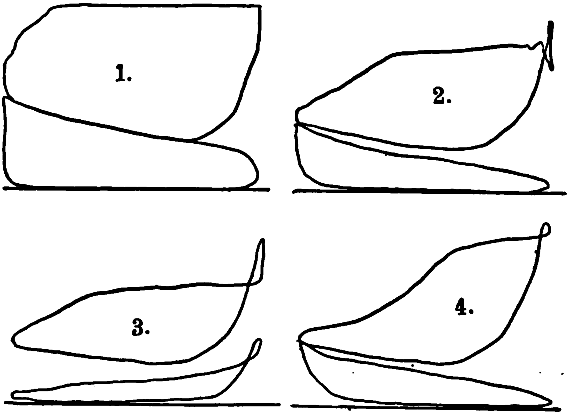

Fig. 58.

Card No.	Rev. per min.	Miles per hour.	Cut-off h. p. cyl.	M. E. P. h. p. cyl.	M. E. P. l. p. cyl.	M. E. P. l. p. ref. h. p.	Horse power.	Per cent. of work by l. p.
1	54	11	20″	100	50.	139	353.8	58
2	218	44	17″	65	27.	75	836.7	54
3	300	60	17″	47	16.	44	753.2	48
4	150	30	9″	58	24.5	68	607.2	54

In making this comparison it has been assumed that the diagrams from the two ends of the cylinders are alike in each case. The diagrams were divided by ordinates as in

Fig. 59, and the difference between the forward pressure on one side of each piston and the back pressure on the other side was plotted for each ordinate. A curve was thus established which represented the net effective pressure on each piston for various piston positions. Then the differences of the ordinates of these two curves gave the curves shown in Fig. 59. The numbers of the curves in this diagram refer to the correspondingly numbered indicator cards of Fig. 58, and the vertical distances above the base line *A*, *A*, represent the excess of the total pressure on the low-pressure piston above that on the high-pressure piston. Distances below the base line mean, of course, that the total pressure upon the high-pressure piston exceeds that on the low-pressure piston. The scale of pressures is about nine thousand pounds to one inch, and the stroke is taken to be in all cases from right to left. It will be seen that the

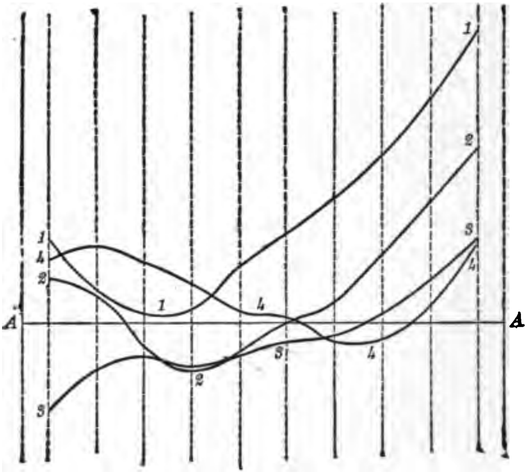

Fig. 59.

greatest difference in pressure is for the diagram taken at slow speed and late cut-off, and that for high speed and early cut-off the difference is comparatively small. Also that, according to curves 2 and 3, the effect of higher speed and lower initial pressure with the same cut-off is to greatly change the amount and distribution of the excess pressure. The scale to which Fig. 59 has been constructed is not sufficiently large to make the results entirely trustworthy, but the curves are sufficient to show that the variation in total pressures at ordinary running speeds is approximately 5,000 pounds. The tendency is to tip the cross-head, and hence to bring an additional bending load on the piston rod of the underloaded piston, which would apparently be most severe where this rod enters the cross-head. It does not follow that this fact is an argument against the adoption of this design, but simply that a varying load of, say, 5,000 pounds acting with a leverage of about 18 inches, and having from 300 to 600 reversals per minute at ordinary speeds, is worth considering, and should be provided for in addition to the usual stresses on piston rods.

The principal dimensions of this locomotive are as follows:

Diameter of h. p. cylinders	12 in.
Diameter of l. p. cylinders	20 in.
Stroke of pistons	24 in.
Ratio of cylinder volumes	2.78
Number of driving wheels	4
Diameter of driving wheels	66 in.
Style of boiler	Wagon top.
Diameter of shell	58 in.
Thickness of shell, steel	½ in.
Size of fire box	length, 108 in.; width, 34 in.
Tubes, 251, 2-in., length	11 ft. 10 in.
Total weight in working order	105,000 lbs.
Total weight on driving wheels	75.000 lbs.
Total wheel base, engine	21 ft. 10 in.
Driving wheel base	7 ft. 6 in.
Total wheel base, engine and tender	50 ft.

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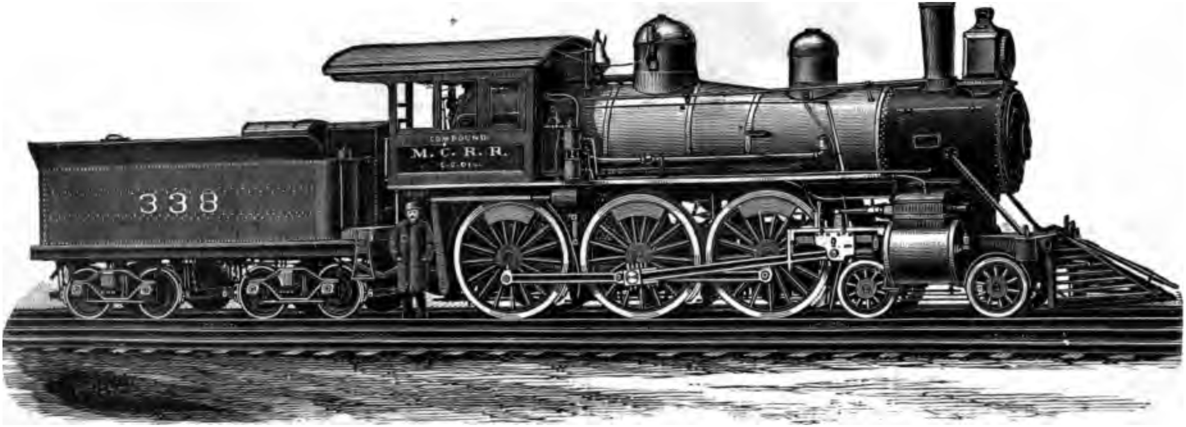

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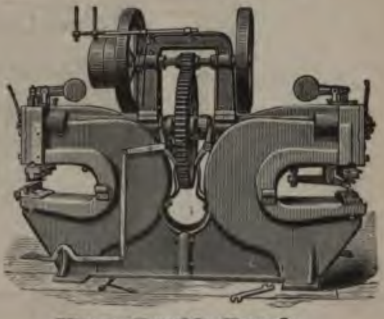

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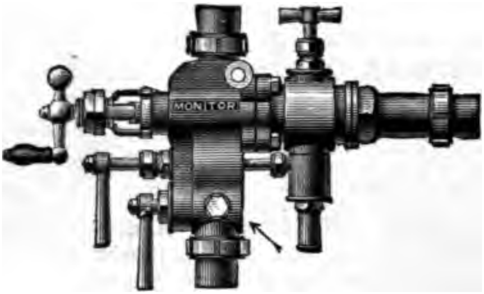

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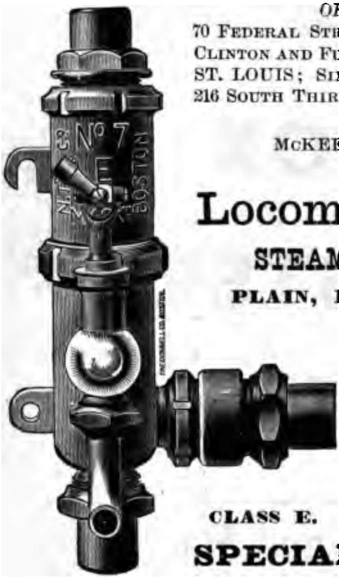

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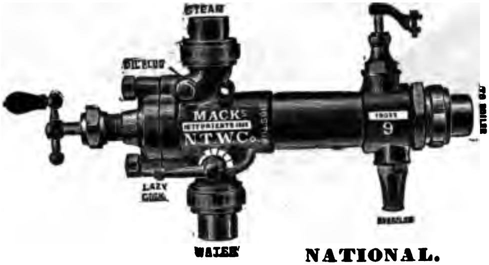

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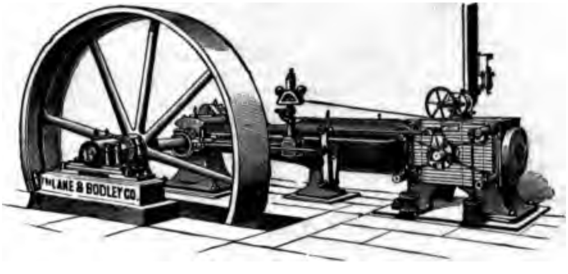

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